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HVAC DESIGN FOR INDOOR GROWING ROOMS

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Abstract: On vertical farms, food is grown in vertical layers in closed, fully controlled environments, where environmental conditions are artificially recreated to increase crop yields. In addition to lighting and irrigation, temperature, humidity and ventilation need to be adjusted according to the needs of each plant. Therefore, it is essential to understand these needs and the biological phenomena of plants to develop HVAC systems that can provide adequate control of temperature, humidity and ventilation. The calculation of the thermal load is the first step in the design phase and this article aims to develop a specific calculation methodology for this type of environment. Definition of temperature and humidity setpoints, generation of sensible and latent heat, influence of lighting and ventilation, strategies for controlling temperature and humidity are the topics covered. In addition, a case study is presented in which the total thermal load for the period with the lighting on was 109.48 kW with a sensible heat factor of 0.476, resulting in a density of 1.00 kW/m². For the period with the lighting off, the thermal load was -3.60 kW (heating) and 6.00 kW of latent heat (dehumidification), representing a sensible heat factor of -1.532. The most suitable system was the chilled water system with water condensation chillers to enable the use of part of the condensation water flow in the heating coils, improving the energy efficiency of the system.
Keywords: HVAC, Design, Temperature, Farm, Indoor.

INTRODUCTION

With the world population growth and the need to produce food in a healthy, sustainable way and without harming the environment, the practice of agriculture indoor in urban farms is an excellent option to meet this demand.In urban farms, food is grown indoors and controlled, where natural conditions are artificially recreated in order to provide the ideal conditions for each type of plant. As it is a controlled environment, in indoor cultivation there is no need to use pesticides and production is not affected by the climatic changes of the seasons. Fig. 1 shows the interior of an urban vertical farm.

While sunlight is replaced by LED lamps, weather conditions are artificially reproduced through air conditioning with temperature, humidity and air speed controls. According to Urban Health Farms (2019) it was in 2001 that the professor at Columbia University in New York, Dickson Despommier, developed the first prototype of an urban vertical farm, so it is a new production method in a global context.

According to Kateman (2020), the indoor agriculture market, which was valued at US$ 23.76 billion in 2016, is expected to reach US$ 40.25 billion in the year 2022. This growth trend and the scarcity of technical literature represent a challenge for the HVAC-R sector, as it is not possible to calculate climate control systems for cultivation facilities with the same methodology used for thermal comfort facilities or commercial refrigeration, since plants are organisms different from human beings, with completely different needs. and therefore require a differentiated approach in the development of projects.

LITERATURE REVIEW

DEFINITION OF INTERNAL CONDITIONS

The air conditioning project starts with the calculation of the thermal load, which in turn depends on the desired temperature and humidity conditions.For thermal comfort, there are standards that can be consulted to define indoor air conditions, but in cultivation facilities indoor the setpoint of temperature and humidity is defined by the Vapor Pressure

Figure 1. Interior of an urban vertical farm.

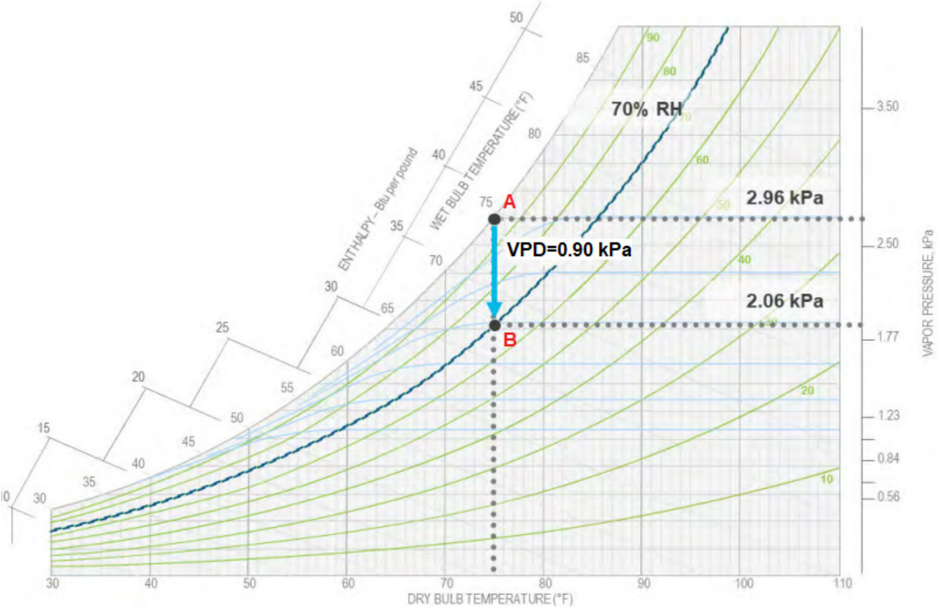

Figure 2. Calculation of VPD. A) Vapor pressure at the surface of the leaves, admitted as saturated air. B) Ambient air conditions.

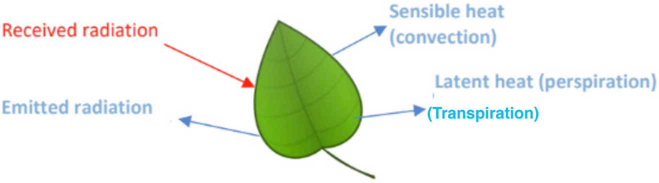

Figure 3. Energy balance on a leaf.

Difference (VPD). The VPD is the difference between the vapor pressure of the air at the surface of the leaves (assumed to be that of saturated air at the leaf temperature) and the vapor pressure of the ambient air, as illustrated in Fig. 2 (TRANE TECHNOLOGIES INC., 2021).

PLANTS: SENSIBLE AND LATENT HEAT

According to Bonan (2016) a leaf absorbs long-wave solar radiation, emits long-wave radiation and exchanges sensible and latent heat with the surrounding air, according to the energy balance shown in Fig. 3.The energy balance in a leaf is given by Eq. (1), where R_n is the net radiation(W/m^2), S_h is sensible heat (W/m^2) and L_h is the latent heat (W/m^2). The radiation emitted by a leaf can be considered negligible in relation to the portions of sensible and latent heat.

$$R_n = S_h + L_h \tag{1}$$

SENSIBLE HEAT

The sensible heat transfer in a plant is dominated by the mechanisms of convection and temperature difference between the leaf and the air. According to Ahmed et al. (2020) the sensible heat flux can be calculated according to Eq. (2).

$$S_h = 2h_c(T_f - T_a) \tag{2}$$

Where h_c is the average convection coefficient (W/m^2K), T_f is the leaf temperature (K) and T_a is the air temperature (K). The h_c can be calculated from the Nusselt number (Nu) according to Eq (3).

$$h_c = k_a \frac{Nu}{L_f} \tag{3}$$

Where k_a is the thermal conductivity of the air at the foil interface (W/m*K) and L_f is the characteristic length of the leaf (m), which can be estimated as the average between the width and the length. According to Incropera et al.(2007) k_a can be estimated by Eq. (4).

$$k_a = (6{,}84x10^{-5})T_b + (5{,}62x10^{-3}) \tag{4}$$

Where T_b is the air temperature at the interface with the leaf (K) and can be calculated as the average temperature between T_a e T_f.

According to Ahmed et al. (2020) Nu can be calculated by Eq. (5).

$$Nu = 0{,}66 \frac{R^1}{2} \frac{Pr^1}{3} \tag{5}$$

Where Re is the Reynolds number and Pr is the Prandt number (0.70 for air). According to Incropera et al. (2007) Re can be calculated by Eq (6).

$$R = \frac{v_a L_f}{\mu_a} \tag{6}$$

Where v_a is the air speed (m/s) and μ_a the kinematic viscosity of air (m^2/s) at the interface with the leaf. μ_a can be calculated by Eq. (7).

$$\mu_a = (9x10^{-8})T_b - (1{,}13x10^{-5}) \tag{7}$$

LATENT HEAT

The transpiration of plants is the process responsible for the greatest load of latent heat in the environment. The latent heat generated by plant transpiration is so high that the Sensible Heat Factor (FCS) can be less than 0.5 (McGowan, 2020), while for thermal comfort this index usually varies between 0.7 and 0.9 (Trane, 2021). In this way, the thermal load of cultivation facilities is dominated by latent heat, unlike thermal comfort facilities that are dominated by sensible heat.

Transpiration plays a critical role in the calculation of the heat load, as it adds moisture to the air without adding energy (isoenthalpic process), in the same way as in evaporative coolers. During this evaporative cooling process, the water absorbs sensible heat from the leaf until it absorbs enough to evaporate, cooling the leaf and adding moisture to the air. As transpiration is a plant's cooling mechanism, the leaf becomes colder than the air and therefore helps to cool the environment.

According to Ahmed et al. (2020) the latent heat flux can be calculated according to Eq. (8).

$$L_h = L_{h,mol} M_w \lambda_E \tag{8}$$

Where L_h is the latent heat flux(W/m²), $L_{h,mol}$ is the transpiration rate in molar units (mol/m²s), M_w is the molar mass of water(kg/mol) e λ_E is the latent heat of vaporization of water $(2{,}26 x 10^6 \frac{J}{kg})$. The $L_{h,mol}$ can be calculated based on the estimate of the total water vapor conductance g_{tv} (m/s) and the difference in water vapor concentration at the leaf surface $C_{w\text{-}f}$ (mol/m³) and in the surrounding air $C_{w\text{-}a}$ (mol/m³), using Eq. (9).

$$L_{h,mol} = g_{tv}(C_{w-f} - C_{w-a}) \tag{9}$$

The g_{tv} is calculated by Eq. (10).

$$g_{tv} = \frac{g_{bv} g_{sv}}{g_{bv} + g_{sv}} \tag{10}$$

Where g_{bv} (m/s) is the foil boundary layer conductance, which is a thin layer of still air that surrounds each foil and g_{sv} (m/s) is the stomatal conductance, used to describe the rate of diffusion of water vapor molecules through the stomata of leaves into the air. The g_{bv} can be calculated by Eq. (11).

$$g_{bv} = \frac{a_s h_c}{\rho_a C_p Le^{2/3}} \tag{11}$$

Where a_s is the transpiration area ratio, being equal to 1.0 for leaves with stomata on both sides, p_a is the density of the air (kg/m³), C_p is the specific heat of air (J/kgK) and is the Lewis number, calculated from Eq. (12).

$$= \frac{\alpha_a}{D_{AB}} \tag{12}$$

Where a_a is the thermal diffusivity of air (m²/s) and is the binary diffusion coefficient of water vapor in air(m²/s). According to Incropera et al. (2007) a_a e D_{AB} can be calculated by Eq (13) and (14), respectively.

$$\alpha_a = (1{,}32x10^{-7})T_b - (1{,}73x10^{-5}) \tag{13}$$

$$D_{AB} = (1{,}49x10^{-7})T_b - (1{,}96x10^{-5}) \tag{14}$$

Aconcentration of water vapor on the leaf surface, $C_{w\text{-}f}$, can be calculated by Eq. (15).

$$C_{w-f} = \frac{P_{w-f}}{R_{mol} T_f} \tag{15}$$

Where $P_{w\text{-}f}$ is the leaf water vapor pressure (Pa), assumed to be the same as that of saturated air at the same leaf temperature and is the universal gas constant $(8{,}314472 \frac{J}{mol K})$.

The concentration of water vapor in the surrounding air, $C_{w\text{-}a}$, can be calculated by Eq. (16).

$$C_{w-a} = \frac{P_{w-a}}{R_{mol} T_a} \tag{16}$$

Where $P_{w\text{-}a}$ is the water vapor pressure of the air (Pa).

According to the study carried out by Kim et al. (2004) stomatal conductance, g_{sv}, of lettuce leaves grown under red and blue LEDs varies between 0.6 and 1.0 mol/m²s.

INFLUENCE OF LIGHTING

In indoor cultivation, sunlight is replaced by artificial lighting specially developed to stimulate photosynthesis. From the thermal point of view, while lighting represents the largest source of sensible heat generation, it is directly linked to the generation of latent heat by transpiration. This is because while the lights are on, the plants receive a greater amount of radiation and this increases their transpiration.

With the lighting turned off, the plants continue to cool the environment because with the ventilation in operation there is still heat transfer by convection in the leaves. On the other hand, latent heat generation is greatly reduced in this period, since without lighting the plants do not receive direct radiation, reducing the need for transpiration as a means of balancing their temperature. With a low transpiration rate, high relative humidity (above 90% for the period without lighting) and good irrigation (since the water circulation system remains in operation), the plants go through the guttation process, where they lose water in the liquid through the leaves, as shown in Fig. 4. With the surface of the leaves wet and with the ventilation in operation, evaporation occurs with a consequent increase in relative humidity, ie, generation of latent heat.

INFLUENCE OF VENTILATION

Along with light, chlorophyll and water, plants need CO2 to produce sugars that provide energy for their growth. Chlorophyll is a substance that exists in all photosynthetic beings, light is provided through artificial lighting, water is provided by irrigation and CO2 is provided through supplementation and good ventilation.

Due to transpiration, the leaves of plants become moist, providing an environment conducive to the development of fungi that cause diseases. Good ventilation on the leaf surface has the drying effect, which prevents the formation of mold while stimulating the leaf to perspire more, accelerating its growth.

From a thermal point of view, ventilation also has the role of homogenizing temperature, humidity and CO2 concentration at all points in the growing environment, so that there are no significant gradients of temperature, humidity and ventilation within the same space. Good ventilation also increases the convective effect on hot surfaces, such as those of electrical equipment that dissipate heat, increasing the rate of heat transfer and preventing overheating of mechanical and electrical components.

Even ventilation provides several benefits in the growing environment, care must be taken with the speed of air that reaches the plants, as speeds above 1.0 m/s can burn the leaves (windburn) and retard growth. The speed must be kept at 0.5 m/s (ASHRAE, 2019).

In thermal comfort installations, ventilation must be maintained at values between 0.05 and 0.2 m/s in the occupancy zone and these values are easily obtained with the correct selection of diffusers. On the other hand, to provide homogeneous speeds of 0.5 m/s in cultivation environments, in addition to the large number of supply points, large networks of ducts and more powerful fans would be required. That is why it is recommended to separate the ventilation into two systems:

- Supply of cold air from the air handling unit;
- Internal air circulation;

Fig. 5 shows an example of separate ventilation systems.

OTHER HEAT SOURCES

Other sources of sensible and latent heat are common in indoor growing environments, such as:

- Peripheral or centrifugal pumps;
- Fans for internal air circulation;
- Drivers of the lighting system;
- People;
- Water mirrors and wet surfaces;
- Infiltrations;
- Automatic process machines;

Figure 4. Guttation process.

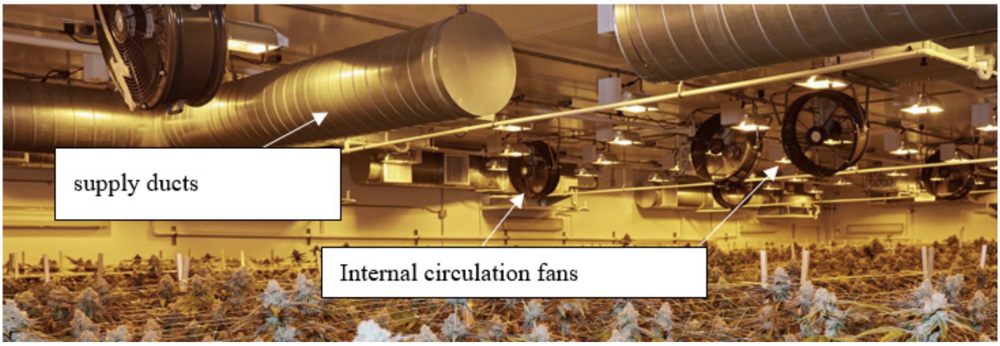

Figure 5. Ventilation systems in an indoor growing environment.

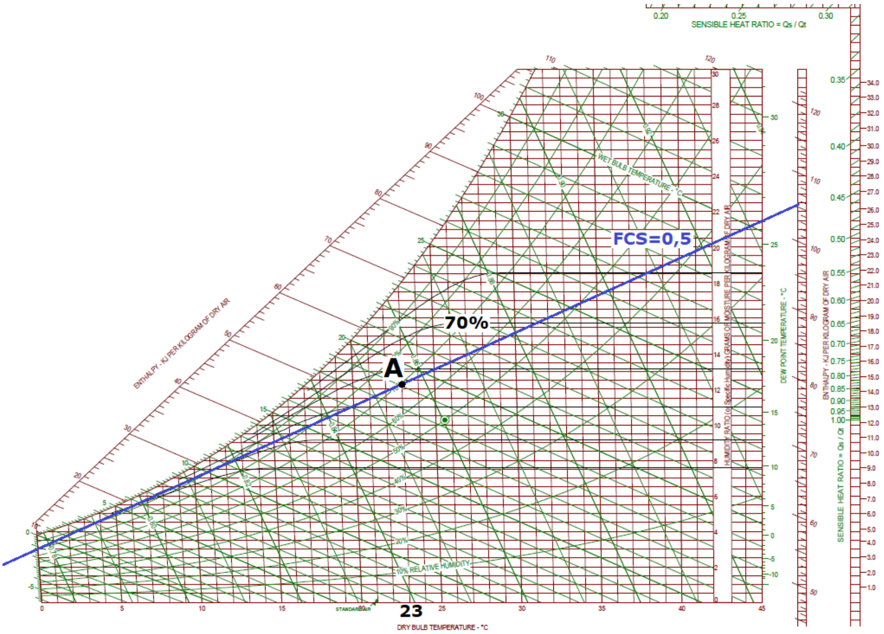

Figure 6. Example of dimensioning with FCS for maximum thermal load.

- Forklifts and/or transport equipment;

Due to the enrichment of CO2 in the cultivation environments, it is not common to renew air because the outside air has low concentrations of CO2, which would represent an additional cost with CO2.

CONTROLLING TEMPERATURE AND HUMIDITY

Among the main climate control systems for temperature and humidity control are the by-pass damper, variable air volume (VAV) and constant air volume (CAV) with reheat. However, systems with by-pass damper and VAV cannot fully control the latent thermal load in partial operating conditions, so the best system to control sensible and latent loads in partial conditions is the CAV with reheat (Trane, 2012). Even though the CAV consumes more energy than the VAV, in this case, because the CAV does not reduce the flow, there is no reduction in the ventilation effect inside the room and this is desirable.

The maximum thermal load occurs when the lighting is on, due to the generation of sensible heat from the LEDs and lighting drivers and of latent heat due to the high transpiration rate of the plants. At this point, it is common for the FCS to remain around 0.5 and the HVAC system must carry out the cooling and dehumidification processes. When dimensioning the air conditioning system to meet the maximum load, infinite state points that are under the FCS line can meet both the sensible and latent load, just by using the appropriate flow for each point. In this case, and only in this case, the temperature and humidity conditions are met only with cooling and dehumidification. In the psychrometric chart in Fig. 6, a thermal load line with FCS = 0.5 is plotted passing through point "A" that represents the conditions of a room with 23 °C and relative humidity (RH) of 70%. Any air state point that is to the left of point "A"above the FCS line will meet the sensible and latent thermal loads, as long as the flow is adequate.

In the period without lighting, practically all the sensible load is eliminated, however, with the ventilation in operation, the plants continue to transpire and generate latent heat at a lower rate, in this way, the effect of evaporative cooling of the air by the plants themselves, which requires heating. and dehumidification.

CASE STUDY

To exemplify the procedure for determining the thermal load and selecting equipment for air conditioning in an indoor cultivation environment, an example will be taken of a cultivation environment with dimensions of 14.5 m in length, 7.5 m in width and ceiling height. of 7 m.

The Tab. 1 shows the setpoints and air properties for the period with and without lighting. The outdoor air values were obtained from the ASHRAE Handbook – Fundamentals (2017) considering the maximums of the last 10 years in the city of São Paulo. On the other hand, indoor air conditions are determined by the needs of the production process.

The Tabs. 2 and 3 present the thermal load results for the periods with and without lighting, respectively. Due to the procedure for calculating the sensible and latent heat plots of the plants, the thermal load was manually calculated using spreadsheets. It is possible to notice that in the period with lighting, cooling and dehumidification are necessary, with an FCS of 0.476. In the period without lighting, due to the evaporative cooling effect of the plants and the generation of latent heat, heating with dehumidification is necessary, with an FCS of -1.532.

For the period with lighting, the total thermal load was 1.00 kW/m², with 0.48 kW/m² of sensible heat and 0.52 kW/m² of latent

	Setpoint with lighting	Daytime Outdoor Air	Setpoint no lighting	Outdoor Night Air	
Dry bulb temperature	23,0	35,7	18,0	6,0	°C
Relative humidity	70,0	49,9	91,5	50	%
Wet bulb temperature	19,0	26,5	17,1	2,0	°C
Dew point	17,2	23,6	16,6	-3,6	°C
Absolute humidity	13,48	20,19	12,93	3,15	g/kg(d.a)
Enthalpy	57,4	87,7	50,9	13,9	kJ/kg
Vapor pressure	1,97	2,92	1,89	0,47	kPa
Density mass	1,0879	1,039	1,1069	1,1613	kg/m³
Specific heat	1,031	1,043	1,030	1,012	kJ/(kgK)

Table 1.Setpoints and air conditions in periods with and without lighting.

Item	Sensible load [kW]	%	Latent load [kW]	%	Total load [kW]	%
Walls and ceiling	7,06	13,5%	0,00	0,0%	2,06	1,9%
Floor	0,06	0,1%	0,00	0,0%	0,06	0,1%
ventilators	14,70	28,2%	0,00	0,0%	4,70	4,3%
Pumps	13,00	24,9%	0,00	0,0%	8,00	7%
People	0,85	1,6%	1,28	2,2%	2,13	1,9%
Lighting	34,75	66,7%	0,00	0,0%	24,75	22,6%
Drivers	8,59	16,5%	0,00	0,0%	38,59	35%
Wet floor	-0,62	-1,2%	0,62	1,1%	0,00	0,0%
infiltrations	0,87	1,7%	1,13	2,0%	2,00	1,8%
plants	-27,14	-52,1%	54,34	94,7%	27,20	24,8%
Total	**52,11**	**100,00%**	**57,4**	**100,0%**	**109,48**	**100,0%**

Table 2. Thermal load – with lighting.

Item	Sensible load [kW]	%	Latent load [kW]	%	Total load [kW]	%
walls and ceiling	-1,94	53,9%	0,00	0,0%	-1,94	-82,5%
Floor	-0,26	7,2%	0,00	0,0%	-0,26	-11,1%
Ventilators	4,70	-130,5%	0,00	0,0%	4,70	199,6%
pumps	8,00	-222%	0,00	0%	8,00	340%
Wet floor	-0,21	5,8%	0,21	3,5%	0,00	0,0%
Infiltrations	-0,30	8,3%	-0,49	-8,3%	-0,79	-33,6%
Plants	-13,59	377,3%	6,24	104,8%	-7,35	312,1%
Total	**-3,60**	**100,00%**	**6,0**	**100,0%**	**2,35**	**100,0%**

Table 3. Thermal load – without lighting.

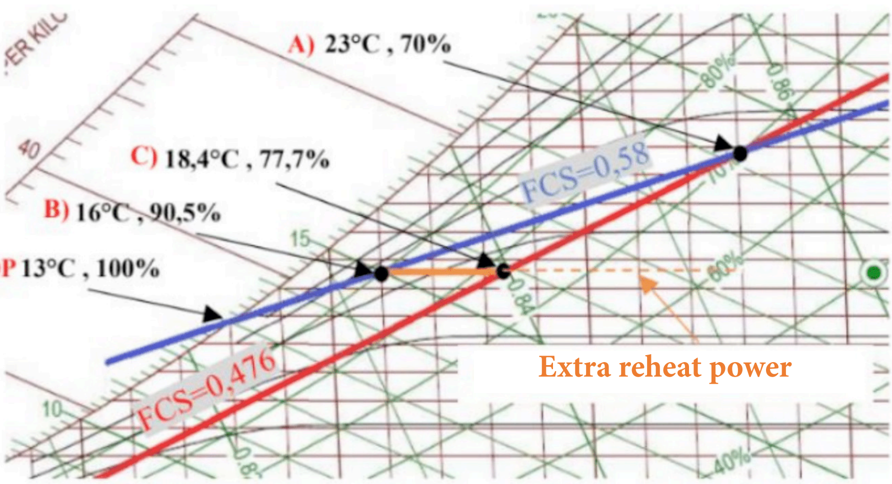

Figure 7. Psychrometric processes for the period with lighting.

heat. For comparative purposes, according to the Ministry of the Environment (2017) a typical value of thermal load ratio per area using as parameters the minimum energy efficiency requirements of ASHRAE 90.1 for a typical commercial building in the city of Atlanta - GA, is of 0.057 kW/m². Thus, it can be observed that the maximum thermal load density of the analyzed cultivation room is about 17.5 times greater than that typical of a commercial building.

Fig. 7 illustrates the psychrometric process for the period with lighting. In it, it is possible to notice that the curve of FCS=0.476, (of the thermal load) that passes through the setpoint does not intercept the saturation curve (relative humidity 100%), indicating that for these conditions there is no coil capable of removing 47.6% of sensible heat and 52.4% of latent heat. For this reason there is a coil curve with FCS arbitrated at 0.58, which passes through the setpoint and which intercepts the saturation curve at approximately 13 °C.This intersection point is commonly called the apparatus dew point (ADP) and is defined as the effective temperature of a coil where cooling and dehumidification takes place.

Thus, a reheating process must be added to the cooling and dehumidification process, because to control the humidity, the air must be blown into the environment at a point that is above the FCS curve of the thermal load (0.476). Thus, air at a flow rate of 11.05 kg/s is initially cooled and dehumidified from 23°C and RH = 70% to 16°C and 90.5%, subsequently reheated from 16°C and 90.5% to 18.4 °C and 77.7% and finally supplied into the environment with the ideal temperature and humidity ratio to absorb sensible and latent loads. The Tab. 4 presents the capabilities of each equipment for the period with lighting. The minus sign indicates the heat removed from the air while the plus sign indicates the heat added to the air. You can see that the total chiller capacity should be 137.099 kW and the reheat coil capacity 27.177 kW. Note that the total process capabilities of Tab. 4 correspond to the thermal loads presented in Tab. 2.

It is important to remember that the above process is the one for the maximum thermal load conditions, however, during the system operation, partial thermal load conditions may occur, such as in winter, where there is a reduction in the thermal load. In these cases, the FCS will be different from the maximum thermal load (0.476) and the supply condition will be different. Therefore, to ensure moisture control under any partial load conditions, it is highly recommended that the reheat coil be sized for an FCS=0, a condition represented by the dashed line in Fig. 7. In this way, the system will be able to meet any partial conditions where the FCS is between 0 and 0.476.

One strategy to reduce the energy spent on reheating is to insert equipment that, in operation, generates sensible heat into the air-conditioned space. If sufficient sensible heat is added to the climate-controlled space so that the FCS of the heat load matches the FCS of the coil, reheat will not be triggered at maximum heat load conditions, being required only at partial loads with reduced FCS.

Fig. 8 illustrates the psychrometric process for the period without lighting, where air at a flow rate of 2.36 kg/s is first heated from 18°C and RH= 91.5% to 25.2°C and 58.9% and then it is cooled and dehumidified up to 19.5 °C and 76.4%, and only then it is supplied into the environment with the ideal conditions of temperature and humidity to absorb sensible and latent loads. The Tab. 5 presents the equipment capabilities for the period without lighting. Note that the total process capabilities of Tab. 5 correspond to the thermal loads presented in Tab. 3.

As can be seen from the temperatures in Fig. 7, especially the ADP, the chilled water

Equipment	Sensible load [kW]	Latent load [kW]	Total load [kW]
Chiller	-79,556	-57,543	-137,099
Reheating	27,177	0	27,177
Process total	**-52,379**	**-57,543**	**-109,922**

Table 4. Capacities of air treatment equipment for the period with lighting.

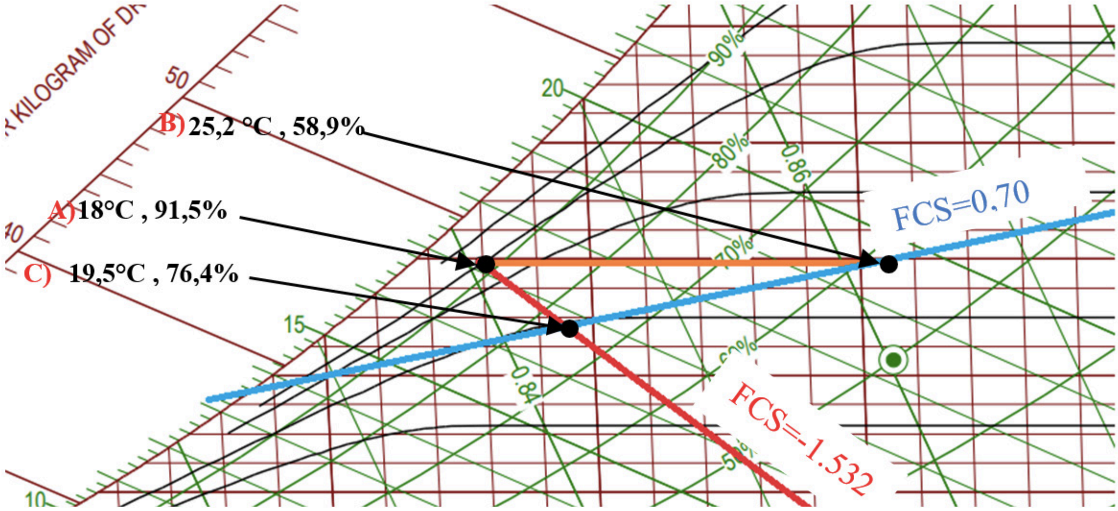

Figure 8. Psychrometric processes for the period without lighting.

Process	Sensible Load [kW]	Latent Load [kW]	Total Load [kW]
Chiller	-13,823	-6,000	-19,823
Cooling and dehumidification	17,461	0,000	17,461
Process total	**3,638**	**-6,000**	**-2,362**

Table 5. Capacities of air treatment equipment for the period without lighting.

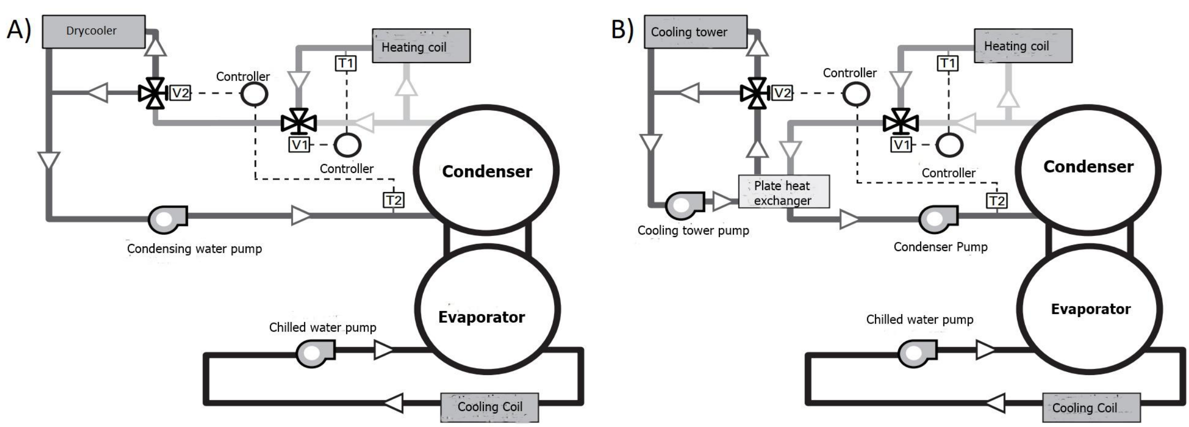

Figure 9. Hot water control flowchart with 3-way valves. A) For drycooler. B) For cooling tower.

coil can operate with conventional supply and return temperatures of 7°C and 12°C, having a total capacity of 137.1 kW with FCS=0.58. Already in Fig. 8, it can be seen that as the cooling process takes place between higher temperatures, from 25.2°C and RH=58.9% to 20.2°C and 70%, to maintain the supply and return temperatures of chilled water coil should have a FCS=0.7 and a total capacity of 19.8 kW. As the cooling and dehumidification load of the period with lighting is about 7 times greater than that of the period without lighting, it is recommended to use different coils for each period, being able to use the same chiller for both.

In this case, it is recommended to use chillers with water condensation, as the heating temperatures shown in Fig. 7 and 8 are close to the inlet and outlet temperatures of the water in the condensers of the chillers (29,4°C a 34,6°C), condensed water can be used as a source of hot water for the heating coils, improving the energy efficiency of the system. For this system it is important to have capacity and flow control for the drycooler (or cooling tower) to prevent the water from returning to a temperature lower than the minimum allowed by the chiller. Fig. 9 illustrates a schematic flowchart of control using 3-way valves for installations with drycoolers and cooling towers.

In Fig. 9 the 3-way valve V2 and the fans of the drycooler (or cooling tower) are controlled according to the temperature measured by the T2 sensor to keep the temperature of the water entering the condenser within the operating limits of the chiller and the heating coil.

The 3-way valve V1 is controlled by the temperature sensor T1 which monitors the water temperature at the return of the reheat coil. An auxiliary heat source may be required if the water temperature at the condenser outlet is not high enough to meet the heating load. If there is hot water generation in the plant, it is recommended to consider in the dimensioning of the boiler sufficient capacity to meet the heating thermal load in case of unavailability of the condensation water of the chiller. Alternatively, finned electrical resistance batteries can be used.

In systems with a cooling tower, the hot water circuit is open to the atmosphere, and it is highly recommended to use a plate heat exchanger (intermediate) between the tower and the chiller condenser to prevent dirt and scale from entering the condensation water circuit. As this system requires the installation of two centrifugal pumps, one dedicated to the tower and the other dedicated to the condenser, whenever possible it is preferable to use drycoolers, as in these equipments the hot water circuit is closed, there is no need for a heat exchanger. intermediate or second pump, eliminating downtime for cleaning the exchanger and improving energy efficiency.

CONCLUSION

The strong dependence on environmental conditions for the production process and the complexity in determining the thermal load due to the little-known nature of the phenomena that contribute to the generation of sensible and latent heat in plants, raise the criticality of the air conditioning project and strengthen the need of a specific calculation methodology for vertical agriculture indoor.

In this article it was possible to understand the main particularities that characterize cultivation environments indoor. With the case study it was found that the air treatment system must be flexible to meet different conditions of temperature and humidity. The total thermal load density was 1.00 kW/m², which is about 17.5 times higher than the typical density for a commercial building sized in accordance with the minimum requirements of ASHRAE 90.1. The thermal load was dominated by latent heat, resulting in a sensible heat factor of 0.47 for the period with lighting on and -1.532 with lighting off.

With the strong approach around the FCS, it is concluded that the cooling and dehumidification coils must be customized to reach, or approach the maximum, the FCS of the thermal load to reduce the need for reheating, which can be carried out with part of the condensation water flow from the chillers. The use of heating coils is essential to control the relative humidity of the air both in conditions of maximum thermal load and in conditions of partial loads.

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