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Artículo Original

Application of building energy simulation in the validating of operational strategies of HVAC systems on a tropical hotel

Aplicación de la simulación energética de edificios en la validación de estrategias operacionales de sistemas HVAC en un hotel tropical

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Abstract

This paper validates a trading strategy of a water-cooled chiller centralized system on a tropical hotel considered a transit hotel during last year 2013 was implemented empirically. For the same thermal loads climate with a building simulation tool called TRNSYS software for critical conditions, occupancy, weather and constructive characteristics of the system was calculated. Thermal load profiles for different levels of occupancy, comparing the installed thermal capacity were evaluated. Also it takes into account the particularities of occupancy in the hotel. This research proposed measures that complement the hotel's energy management. Finally the strategy implemented allowed the hotel, savings403 123,76 kWh/year, and issuing leaving 371,27 t of CO2/year into the atmosphere. It meant about average consumption of the whole hotel for three months. It had an economic impact of 53212,33 USD/year.

Key words: operational strategies, HVAC, thermal Load; water-cooled centralized system, chiller.

Resumen

El presente trabajo valida una estrategia de operación del sistema centralizado de agua helada en un hotel tropical considerado un hotel de tránsito que durante el año 2013 se puso en práctica de forma empírica. Para el mismo se calculó las cargas térmicas de climatización con utilizando una herramienta para la simulación de edificaciones, software TRNSYS, para condiciones críticas, de ocupación, meteorológicas y las características constructivas de la instalación. Se evaluaron los perfiles de carga térmica para distintos niveles de ocupación, comparándose con la capacidad térmica instalada. Además se tienen en cuenta las particularidades de ocupación del hotel. En esta investigación se proponen medidas que complementan la administración de la energía. Finalmente la implementación de esta estrategia permitió el ahorro de 403 123,76 kWh /año, dejándose de emitir 371,27 t de CO2/año a la atmosfera. Este ahorro representa el consumo del hotel en un periodo de tres meses y tiene un impacto económico de of 53212,33 USD/año.

Palabras claves: estrategia operacional, HVAC, cargas térmicas, sistema centralizado de agua helada, enfriadoras.

Introduction

In tropical weather country like Cuba, the demand for cooling of indoor air is growing due to increasing comfort expectations and increasing cooling loads. However, Heating, Ventilating and Air-Conditioning (HVAC) systems in hotel facilities are the dominant energy consuming appliances, recent studies reveals that they may represent a 60 % of the total electricity consumption in comparison to other electrical appliances due the high temperature and moisture of the Caribbean zone [1]

HVAC systems used in hotels are usually water-cooled chiller system [2]. The initial investment in these centralized systems is about 2,2 times larger than Air Handling Unit (AHU), but achieved comfort is superior because of its quieter operation, and from an aesthetic point of view inside and outside the building. Chillers present strong opportunities for energy reduction. The energy efficiency of these systems depends on the heavily on its operational control and operational strategies are applied, taking into consideration all the factors involved to achieve minimum energy consumption [3]. Without sacrifice of thermal comfort, it's necessary to reset the suitable operating parameters, such as the chilled water temperature and supply air temperature in order to have energy saving with immediate effect.

Budaiwi and Abdou [4] investigated the impact of operational zoning and HVAC system intermittent operation strategies on the energy performance of mosques, the places of worship for Muslims, while thermal comfort is maintained. Energy simulation modeling was used for evaluating alternative zoning and HVAC operation strategies. Results indicated that up to 23 % reduction in annual cooling energy was achieved by employing suitable HVAC operation strategy and system over-sizing, and 30 % reduction was achieved by appropriate operational zoning. Comparing the cooling energy consumption of HVAC during summer continuous operation of an un-insulated mosque with the consumption of the insulated mosque with properly oversized HVAC system operated for 1 h during each prayer, indicated that as much as 46 % of cooling energy reduction can be achieved. Furthermore, using proper operational zoning and HVAC operation strategies was expected to bring about an additional significant energy reduction.

Ardehali and Smith [5] analyzed various operational strategies applied to older- and newer-type commercial office buildings on Des Moines Iowa, using constant-air-volume-reheat and variable-air-volume-reheat HVAC systems, respectively. The operational strategies were: night purge (NP), fan optimum start and stop (OSS), condenser water reset (CWR) and chilled water reset (CHWR). The results show that, in general, NP is not an effective strategy in buildings with low thermal mass storage, OSS reduces fan energy, and CWR and CHWR can be effective for chillers with multi-stage unloading characteristics. The most energy-efficient operational strategies were the combination of OSS, CWR, and CHWR for the older-type building, and OSS for the newer-type building. Economically, the most effective was the OSS strategy for the older-type building and the CHWR strategy for the newer-type building.

Fong *et al*. [6] presented a simulation-optimization approach for the effective energy management of HVAC system. Using a metaheuristic simulation–EP (evolutionary programming) they suggest optimum settings (chilled water supply and air temperatures) for different operations in response to the dynamic cooling loads and changing weather conditions throughout a year of a local project. This reset scheme would have a saving potential of about 7 % compared to the existing operational settings.

Yu and Chang [7] evaluate operating cost savings of a chiller system integrated with optimal control of cooling towers and condenser water pumps. A sophisticated chiller system model was used to establish how different control methods influence the annual electricity and water consumption of chillers operating for the cooling load profile of a reference hotel. It was estimated that applying load-based speed control to the cooling tower fans and condenser water pumps could reduce the annual electricity consumption on 8,6% and operating cost on 9,9 % relative to the equivalent system using constant speed fans and pumps with a fixed set point of 29,4 ^{o}C for cooling water temperature control.

Starting in 2013, at a hotel in Cuba, taking into account the characteristics of occupation it wasbegun empirically anoperational strategy HVAC system in order to reduce their energy consumption. The purpose of this paper are validate the operation strategy by analyzing thermal load profiles taking into consideration the weather conditions, building characteristics and hotel occupancy, and proposing measures to improve this strategy.

Methods and Materials

Thermal loads methods for facilities

Selecting the appropriate size of a chiller is significant to an efficient chilled water system. Sizing of the chiller or chillers should always be done using a thorough calculation of the maximum space cooling loads and process loads. Loads for space cooling are calculated based on outdoor design conditions, solar loads, estimated cooling loads associated with internal loads from people and equipment, and infiltration and ventilation loads. Once the maximum cooling loads are determined the total size, in tons, of the chiller can be determined.

To guaranteethe thermal loads calculations there are several methods: Instantaneous loads, E20 Carrier and CLTD/ASHRAE CLF, Transfer Functions Method (TFM) and Thermal Balance. Currently simulating thermal loads is one of the most widespread resources available for the determination of thermal loads in buildings. It is supported by integrated software package methodologies. These programs can reduce the execution time of tasks and analyze a large number of possible solutions with minimal resources. For these applications on the market there are a set of packages for thermal simulation of buildings, such as TRNSYS, DOE-2, ENERGY PLUS COOL PACK simulator UABC, Trane TRACE, DP-AIR (Polytechnic University of Valencia), Saunier Duval, MC4, among others.

TRNSYS is a transient simulation program systems with modular structure designed to solve complex energy systems splitting problems into a series of small components and configured assemblies using an integrated graphical interface known as TRNSYS Simulation Studio, where the user specifies the components that constitute the system and the manner in which they are connected. The compiler solves the system of differential and algebraic equations representing the whole system [8]. TRNSYS is compatible with the requirements of ANSI / ASHRAE Standard 140-2001 (ANSI / ASHRAE 2001).

The TRNSYS interface interacts with the user as a graphic programming tool. This means that no previous knowledge of a programming language is necessary to create and run a simulation, although TRNSYS allows modifying its component models with several common programming languages. Component models refer to subroutines that the TRNSYS program libraries incorporate in its standard version, each subroutine models a specific component. The accuracy of a thermal simulation result is determined by the input data. This input data mainly consists of the building geometry, internal loads, HVAC systems and components, weather data, operating strategies and schedules, and simulation specific parameters, as show the figure 1.

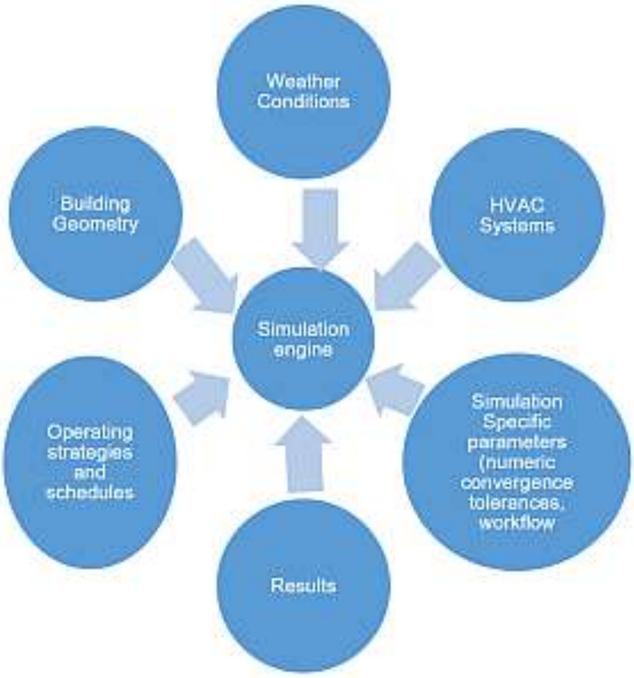


Fig. 1. General input data of thermal simulation engines [9]

TRNSYS is able to determine the dynamics of involvement in the thermal zone due to the following disturbances: temperature of dry and wet bulb, solar radiation, natural and artificial lighting, infiltration, ventilation, occupancy and equipment. For building simulation, type 56, this component models the thermal behavior of building having multiple thermal zones. The building description is ready by this component from a set of external files. The files can be generated based on user supplied information by running the preprocessor program called TRNBuild. The building thermal and simulations model was developed based on the following [10]:

- Building construction detail and occupancy schedules
- List of indoor lights, equipment and machines
- Centralized HVAC systems in the building
- Cooling load types in the building

Heat balance method is used by TRNSYS as a base for all calculations. For conductive heat gain at the surface on each wall, TRNSYS (manual volume 6) use TFM method as a simplification of the complicated heat balance method. The methodology exposed for Bhaskoro *et al*, are, equations 1 and 2: [11]

$$q_{s,i} = \sum_{k=0}^{nbs} b_s^k T_{s,o}^k - \sum_{k=0}^{ncs} c_s^k T_{s,i}^k - \sum_{k=0}^{ncs} d_s^k q_{s,i}^k \tag{1}$$

$$q_{s,o} = \sum_{k=0}^{nbs} a_s^k T_{s,o}^k - \sum_{k=0}^{ncs} b_s^k T_{s,i}^k - \sum_{k=0}^{ncs} d_s^k q_{s,o}^k \tag{2}$$

Where $q_{si/so}$ are the conduction heat flux from the wall at the inside/outside surface. $T_{s,i/s,o}$ the inside/outside surface temperature. These time series equations in terms of surface temperatures and heat fluxes are evaluated at equal time intervals. The superscript k refers to the term in the time series. The current time is k = 0, the previous time is for k = 1, etc. The timebase on which these calculations are based is specified by the user within the TRNBUILD description. The coefficients of the time series (a's,b's, c's, and d's) are determined within the TRNBUILD program using the z-transfer function routines.

Heat gain through radiation and convection within the zone were calculated using the star network given by, equation 3:

$$q_{comb,s,i} = q_{c,s,i} + q_{r,s,i} = \frac{1}{R_{equiv,i} * A_{s,i}} (T_{s,i} - T_{star}) \tag{3}$$

Where R_{equiv} is equivalent resistant between the wall with a node, T_{star}: artificial temperature node, $q_{c,s,I}$ convection heat flux on the internal surface of the walland $q_{r,s,I}$ are the long wave radiation heat flux on the internal surface of the wall. Heat gain through radiation and convection for external surface were calculated by, equations 4, 5 and 6:

$$q_{comb,s,i} = q_{c,s,o} + q_{r,s,o} \quad (4)$$

$$q_{c,s,o} = h_{conv,s,o}(T_{a,s} - T_{s,o}) \quad (5)$$

$$q_{r,s,o} = \sigma\varepsilon_{s,o}(T_{s,o}^4 - T_{fsky}^4) \quad (6)$$

Where, $q_{comb,s,i/o}$ is combined convective and long wave radiation of inside/outside surface. $h_{conv,s,o}$, is the convective heat transfer coefficient on the surface of the wall, Ta,s, is the ambient temperature , T_{fsky}, fictive sky temperature. The Stephan–Boltzmann constant is describe by σ and ε_s is the long-wave emissivity of the surface. Then, total heat gain through inside and outside surface of the wall are, equations 7 and 8:

$$qs,i = q_{comb,si} + S_{s,i} + Wall - gaing \quad (7)$$

$$q_{s,o} = q_{comb,so} + S_{s,i}o \quad (8)$$

$q_{s,i/s,o}$ are surface inside/outside conduction heat flux of the wall,$S_{s,i/s,o}$ the radiation inside/outsideheat flux absorbed on the surface of the wall (solar and radiative gains) and Wall-gain is an user-defined energy flow to the inside wall or window surfaces.

Long wave emissivity was 0,9 for wall. The value based on window library. Solar absortance coefficient for wall based on the table provide by TRNSYS. Convective heat Transfers coefficient for inside and outside wall were set 11 $kJh^{-1}m^{-2}$ and 6411 $kJh^{-1}m^{-2}$ as recommended by the software [12]. Latent and sensible heat gain from ventilation and infiltration air is calculated using, equations 9 and 10 [13]

$$q_{sensible} = m_a Cp(T_o - T_i) \quad (9)$$

$$q_{latent} = V_a \rho_a (\omega_o - \omega_r) h_{fg,32} \quad (10)$$

Where m_a is the mass of air, Cp the specific heat of air, Va the flow rate, ρ the density of outdoor air, ω, humidity and $h_{fg,32}$ is the latent heat of vaporization at 32 ^oF. Minimum ventilation rate required in each room is calculated based on ASHRAE standard present in the equation 11:

$$V\min = R_p * N_p + R_a * A \quad (11)$$

On this equation 11 R_p is the occupant ventilation component, Na, the number of occupant, Ra the building ventilation component and A is the room area.

Other Building simulations assumptions were:

- Heat gain from the electrics device, heat gain from lighting, heat gain from building envelope would contributed to sensible cooling load.
- Heat gain from occupant, ventilation and infiltration would contribute to both latent and sensible cooling load
- The amount of heat gain per occupant is based on ISO 7730 table. Degree level of activity was inputted to get the portion of sensible and latent heat from the table.
- Convective and radiative fraction for heat gain from electric devices were 0,7 and 0,32 while for artificial lights, the values were 0,6 and 0,4 [14]
- Design temperature and RH were based on ASHRAE where 25 ^oC of operative temperature and 50 % of RH for comfort zone
- Infiltration by air change rate and Ventilation supplied by the Air Handing Unit (AHU) as required by ASHRAE standard, were based on type of thermal zone.
- For a major analysis, the simulation was separate into two periods: summer periods and winter periods

Results and Discussion

Main characteristic of the building. Thermal loads

The building considered in the study is a hotel with 149 rooms available for tourism. The main building has a capacity of one hundred thirty six rooms (136), and also features a block of thirteen (13) cabins located in the pool area. It is constructed with a concrete structure with exterior walls double block of six inches, an air space in the center for these walls are acoustic and interior walls of single block, with glass windows in the main building overlooking the northern part and wooden doors in the south aisle. The cabins are constructed in a similar manner except that south wall is made of glass. Shop and restaurant also have very large glazed areas with large heat gains in this area.

The HVAC systems of the hotel are two water chillers model: CHAWT-1402-AT-BP-RC-100 C1-VT, the cooling capacity: 404 kW each. (115 TR) chilled water flow GW-70 m^3/h. Temperature Input / Output: 11/6 °C. To calculate the thermal loads hotel, first it was divided by type rooms. They are classified according to their dimension, neighboring rooms, and the influence of solar radiation on the roof and external walls. Based on these criteria, 13 types of heating zones or rooms with very similar characteristics of thermal load were identified, as shown at figure 2.

Fig. 2. Hotel buildings types zones. Source: Adapted from [15]

All thermal zones were modeled in TRNSYS. In this simulation environment, the thermal load of the resulting guestroom module is calculated as a function of: the comfort set point temperature, weather variables for the situated geographic zone, and the heat gains due to occupancy and equipment presented in the guestrooms. The physical description of the defined thermal zones is shown in table 1 and figure 3.

Table 1. Physical description of thermal zones

Thermal zone (Type rooms)	dimensions (m x m x m)	orientation façade (windows)	Thermal zone (Type rooms)	dimensions (m x m x m)	Orientation faça de (windows)
room 702	9 x 8 x 3	north	room 424	9 x 4 x 3	north
room 602	9 x 8 x 3	north	room 202	9 x 4 x 3	north
room 712	9 x 4 x 3	north	room 212	9 x 4 x 3	north
room 724	9 x 4 x 3	north	room 224	9 x 4 x 3	north
room 512	9 x 4 x 3	north	room 102	7,8 x 4.2 x3,4	south
room 402	9 x 4 x 3	north	room 112	7,8 x 4.2 x3,4	south
room 114	7,8 x 4,2 x 3,4	south			

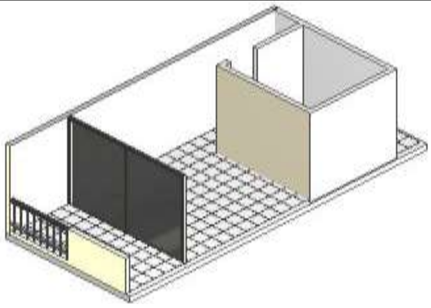


Fig. 3. Top scheme of the room. Source: Authors

The building was simulated in TRNSYS environment, using the TRYNBUILD application, including in TRNSYS package. Four different walls were considered, whose values of transmittance and front and back solar absorbance are shown in table 2.

Table 2. Characteristic of building wall

Wall Type	Material	Thermal conductivity (W/mK)	Overall transmittance (W/m^2K)	Thickness (m)	Solar absorptance of wall		Convective heat transfer coefficient of wall	
					Front	Back	Front	Back
Outwall	Concrete block Cement + Clay Cement + Clay	2,05954	11,40879	0,15 0,02 0,01	0,5	0,3	64	11
Intwall	Brick Cement + Clay Cement + Clay	2,09029	11,67302	0,15 0,01 0,01	0,3	0,3	11	11
Ground	Concrete Ceramics Cement + Clay	3,40909	29,18919	0,24 0,01 0,01	0,3	0,3	11	11
Roof	Concrete Rasilla	3,25960	26,31854	0,24 0,02	0,75	0,3	64	11
Window	Single crystal	5,8	5400	0,008	-	-	64	11

In Cuba the predominant climate is warm and tropical with a rainy season in summer. Its geographical position located near the Tropic of Cancer makes achieving high solar radiation values averaging 5 kWh/m^{2}day. Temperatures are generally high, with average temperatures of 26 ^{0}C and 32 ^{0}C held in the summer, determining the character of its warm climate [16]. Weather variations affect a building, its thermal state, and its technical systems in a variety of ways. This study focused on the effects of the following weather variables: global and diffuse solar radiation and dry bulb air temperature as climate disturbances to the guestroom module dynamics. These data were provided by Cienfuegos´s Weather Institute.

Regarding the heat gains due to occupancy and equipment presented in the guestroom module, it was considered that the internal heat gains were defined for constant occupancy of three people in the room, (100 % occupancy). Profits are also defined by artificial light and a color TV 19-21 Pot = 120 W. For earnings infiltration factor 0,8 was assumed. Under these requirements in table 3 thermal loads for each room type at 24 h are presented. These thermal gains are automatically calculated and added to the total thermal load based on the corresponding values for these magnitudes in TRNSYS according to a hotel.

Table 3. Thermal load of room type. Source: Authors

Places	Room Type	Peak Load (Ton/kW)	Places	Room Type	Peak Load (Ton/kW)
Main Building	No 724	1 /3,52	Cabin	Est side No 114	1,2 /4,22
	No 702	1,5 /5,28		West side No 101	1,4 /4,93
	No 602	1 /3,52		Cabin inside No 107	1 /3,52
	No 712	0,9 /3,17	Restaurant	Restaurant	32,73 /114,8
	No 514	0,4 /1,41	Cabaret	Cabaret	69,99 /245,6
	No 401	0,7 /2,46	Office	Office	40 /142
	No 424	0,6 /2,11	Gift shop	shop	4,5 /15,7
	No 201	0,8 /2,82			
	No 212	0,4 /1,41			
	No 224	0,5 /1,76			

The main building thermal profiles present a dynamic behavior for each hour of the day. figure 4 shows the profile of thermal load for several days in July.

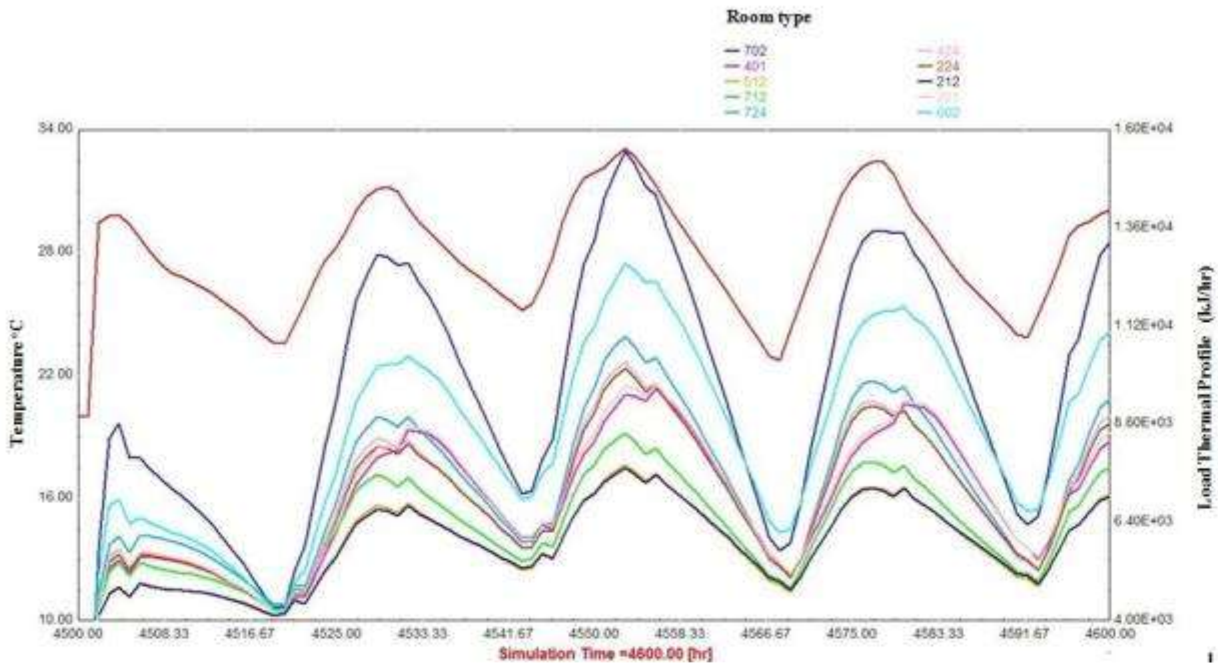


Fig. 4. Load thermal profiles of room type. Source: Authors

Relationship between thermal loading of each cooling circuit and capacity of installed equipment

The HVAC system is composed by 6 hydraulic circuits: the residential block located in the main building is divided into two separate circuits, eastside and the other to the west side; circuit cabins, restaurant circuit, cabaret, and finally, areas nobles' circuit belonging to the offices and shop.

Considering the total room type that integrate each of the hydraulic circuits, the following occupation strategies implemented at the hotel are: first occupy the west side of the main building, from top to bottom, and from left to right, this side of the hotel have a better view of the city. Once occupied this side, the east side deals from top to bottom and left to right. As last alternative the cabins are occupied. In all cases closely meets the requirements of tourists, which usually allows implementing this strategy. The occupation of the two suites is independent.

The values of occupation with the above considerations are: 60 rooms (40 % occupancy), spaced on the west side.89 rooms (60 %), located throughout the westside and in the upper floors from the eastside.119 rooms (80%), located on the west side and in almost the entire east side.149 rooms (100 %), hotel full occupied. figure 5 shows the load profile for each per cent of occupancyunder critical conditions during summer season.

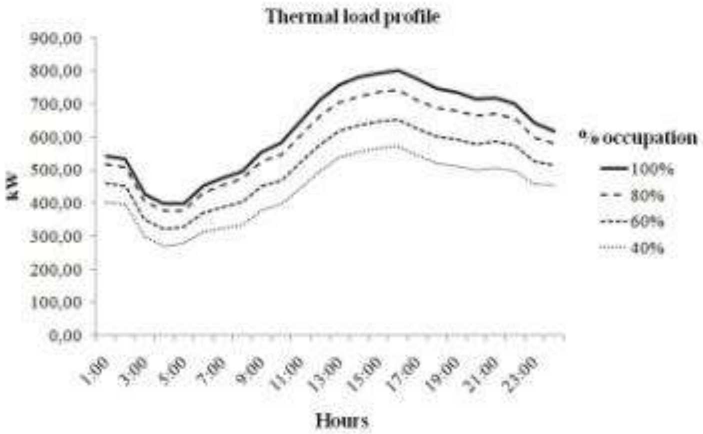


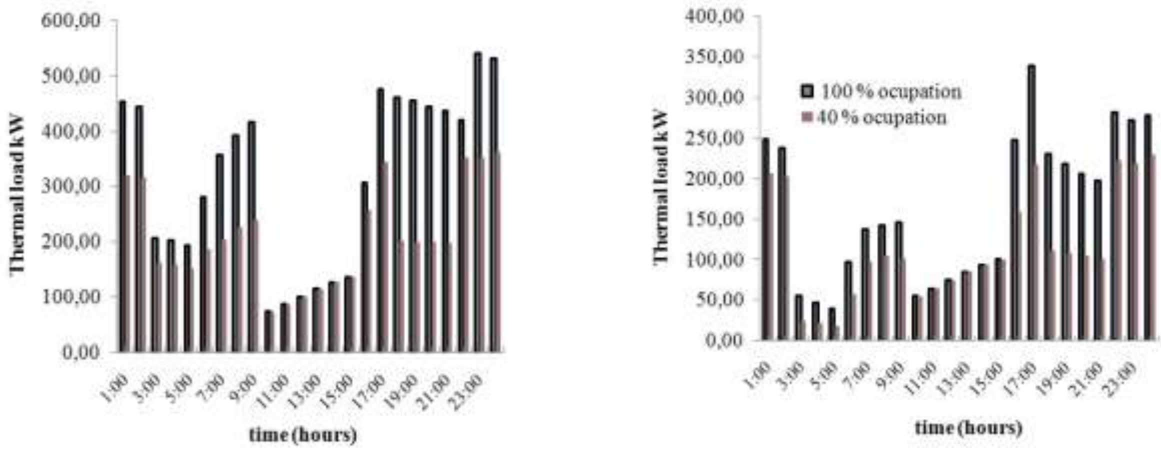
Fig. 5: Thermal load´s profile of the system HVAC at different occupation levels for a day. Source: Authors

The HVAC system meets the demand of the installed load: the installed chiller capacity: 808 kW (230 tons of cooling) and the demand for air-conditioned premises is of 805,17 kW (229,05 tons).

Trading strategies considering hotel occupancy

As described above Jagua hotel is considered a transit hotel. In it, the tourists leave the facility after breakfast to carry out different activities until they return again at 4:00 pm. For these cases the operating strategy of the HVAC system to reduce power consumption without affecting comfort is as follows:

The pump restaurant circuit is connected from 6.30 am to 10.00 am. At 9.00 am rooms circuits (pumps 4,5 and 6) and cabins stop, until 4:30 pm that connects again. Then at 5:30 pm the restaurant circuit is connected again until 10:00 pm. The cabaret circuit is connected from 10:00 pm until 2:00 am. It stays connected throughout the daytime the areas nobles circuits until 5:00 pm. In figure 6a fluctuating loads for occupancy levels corresponding to 100 % and 40 %, ona summer day is displayed, and figure 6b fluctuating loads to the same levels shown, but on a winter day.



a) On a summer day b) On a winter day

Fig. 6: Thermal loads Vs. Occupations' profiles. Source: Authors

Given this behavior of loads, this operational strategy allows:

- Working with a chiller in the hours between 8:00 am and 4:00 pm; with a temperature of ice water (set point) of 9 °C.
- Working with a chiller during the winter months.
- Continue with a chiller if the occupancy is less than 44% between the hours of 4:00 pm and 10:00 pm then connect the other chiller for the cabaret circuit until 2:00 am.

- If the occupancy is less than 44 % occupation as follows: from top to bottom and left to right, leaving latter occupying the top floor and the east corner. With 8 °C chilled water output.
- Working with a chiller in the schedule from 2:00 a.m. to 6:30 am, with set point of 10 °C
- No match schedules service restaurant with cabaret
- Occupy the east side of the building from top to bottom and left to right, leaving as a last resort rooms upstairs and the east corner. In the cabins occupy the middle and east first, leaving the West Side as a last option.
- Working with two different points for every season: in summer and in winter 8/9 ºC to 9/10 ºC.

Total savings that were obtained by applying the new operating strategy and Jagua hotel occupancy was 403 123,76 kWh/year, and issuing leaving 371.27 t of CO_2/year into the atmosphere. It meant about average consumption of the whole hotel for three months. It had an economic impact of 53212,33 USD/year. Furthermore, considering the installed chiller part load work during the daytime, the installation of a smaller capacity chiller is proposed.

Conclusions

The validity of the operating strategy of centralized chilled water system that uses the hotel through a careful calculation of thermal loads throw TRNSYS software, considering the structural characteristics of this installation and weather conditions where it is located was verified. The HVAC system is properly sized. But the performance characteristics of this hotel let take various measures that lead to a considerable saving of electricity, operating costs and environmental impact.

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