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PREDICTION METHOD FOR THE PERFORMANCE OF A CHILLER FOLLOWING A COOLING LOAD PROFILE

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Abstract. *In this paper a method to predict the performance of an air cooled reciprocating compressor chiller is developed. A procedure to select and analyze the cooling capacity of equipments, developed by a manufacturer, is adapted in a computational routine to obtain an energetic assessment of the chiller operation as a function of a cooling load profile and outside air dry bulb temperature profile during a day of operation. The system modeling looks for energy analysis that can be an important information in the decision-making process. The analysis revealed a coefficient of performance between 2.48 to 3.78, an electrical energy consumption close to 2290 kWh per day, and a mean ratio of energy consumption and refrigeration load of 1.196 kW per Ton. The results depend on the number of chiller compressors that are in operation, the cooling load and the ambient temperature, and they promote the use of equipment simulation as a method to evaluate chiller system performance and energy consumption in buildings.*

Keywords: *simulation, vapor-compression, chiller, cooling load, coefficient of performance and control system.*

NOMENCLATURE

$C_{p,w}$	Specific heat of water (kJ/kg.K)
COP	Coefficient of Performance
DBT	Dry bulb temperature
itd/TD	Initial temperature difference fraction at the evaporator
IDT	Initial temperature difference at the condenser
$\dot{m}$	Chilled water flow rate (m^3/h)
$\dot{Q}$	Refrigeration capacity (kW)
$\dot{Q}_n$	Chiller nominal capacity (kW)
$\dot{Q}_R$	Cooling demand (kW)
T_{cd}	Condensation temperature ($^\circ$C)
T_{ew}	Entering chilled water temperature ($^\circ$C)
T_{lw}	Leaving chilled water temperature ($^\circ$C)
T_{ev}	Evaporating temperature ($^\circ$C)
ΔT	Design-chilled water temperature drop ($^\circ$C)
ΔT_R	Operating-Chilled water temperature drop ($^\circ$C)
$\dot{W}_c$	Compressor electricity consumption (kW)
$\dot{W}_f$	Fan electricity consumption (kW)

1. INTRODUCTION

Energy consumption in buildings can represent about the third of the energy consumption of a country, being a concern and principal issue of studies and recommendations that look for its reduction (ASHRAE, 2004).

In order to use less energy in both construction and operation of buildings, numerous countries/regions have developed green building programs aimed at promoting more sustainable buildings. One such program is LEED (Leadership in Energy and Environmental Design) that supporst an integrated design approach (Newsham *et al.*, 2009).

The air conditioning system is frequently responsible by a significant portion of building energy consumption. The reduction of the electricity consumption in buildings can be done by: i) turning lower the building-cooling load (changes in the building structure), and ii) using energy efficiency systems and equipments.

Most of air condition systems operates according a vapor-compression refrigeration cycle. An air cooled refrigeration equipment operates following the cooling loads and dry bulb temperature that occur during a day of operation. In most of the operating hours of an air conditioning system, the equipment operates in part load.

Other chiller optimization options are analyzed in different papers. Aprea *et al.* (2009) identify for variable speed compressors, the current frequency that optimizes the energy, exergy and economy aspects, because not always decreasing compressor speed results in an energy saving when compared to the thermostatic control.

Computer simulation models eliminate the lack of appreciation of the way building respond to the weather and the various environmental and design factors. Therefore, it has become a helping tool for building designers and energy analysts (Fasiuddin *et al.*, 2010). The U. S. Department of Energy (DOE) has developed during the last decade the EnergyPlus building energy simulation software that contributes in the selection and size of air conditioning system and equipment choice (DOE, 2010).

The TRANE company is one of the chiller manufacturers that promotes the use of building simulation as a method of reviewing chiller system enhancements. In a paper, this company applies building simulation to analyze three optimized schemes, which involve: lowering cooling tower temperature to increase chiller efficiency, decoupling the chillers from the chilled water distribution loop, and using unequally sized chiller combinations (TRANE, 1989).

Thermodynamic models and simulations programs were used in predicting the steady state performance of chillers. One model based in the variation of evaporator capacity by changing the mass-flow rate of the refrigerant, while the inlet chilled-water temperature keeps constant, demonstrates that is a useful tool for design and performance evaluation of the refrigeration system (Khan and Zubair, 1999).

Yu and Chan (2009) prove that using variable speed chiller system with load-based speed control could reduce the annual total electricity consume by 19.7% and annual water consume by 15.9% when compared with a typical constant speed system. The thermodynamic chiller system model was developed to locate the optimal operation region with maximum COP.

Chowdhury *et al.* (2009) present the modeling and analysis of air-cooled chiller system in an office building. The paper reviews various thermodynamic simulation models that can be used to predict the performance of a reciprocating air-cooled chiller over a range of operating condition. In addition, the implementation of a passive cooling system could promove 187 kW/m^2/month of cooling energy saving.

Avgelis and Papadopoulos (2009) developed a model that allows the classification of alternative technical solutions concerning the HVAC's design. The HVAC (heating, ventilation and air conditioning) systems are modeled using two simulation tools, in that way the energy consumption is calculated and compared in order to indicate which technical alternative saves more energy.

Beyond thermodynamic model, mathematical models are developed to optimize chiller systems. The particle swarm algorithm is used to minimize energy consumption in multi-chiller loading (Lee and Lin, 2009). Chlela *et al.* (2009) set up a methodology to perform parametric studies in order to design low energy building; this method permits compute mathematical models with a limited number of simulations, reducing the difficulties of modeling complex systems.

In this paper a method looks to predict the system's efficiency based in a manufacturer methodology; this is the selection procedure for air-cooled equipments called Special Mix-Match Selections (1980). This procedure was used considering a chiller with a nominal refrigeration load of 700 kW (200 TR), selected to attend the building peak load.

The thermodynamic procedure allows to study an operational configuration for the refrigeration system. This configuration is compounded by the quantity of compressors that should be turn-on in each refrigeration circuit and their time operation, in order to attend the cooling demands with highest Coefficient of Performance (COP).

2. SYSTEM DESCRIPTION AND ASSUMPTIONS MADE

The project will consider a vapor-compression refrigeration system. The equipment size depends of the cooling load profile based on a critical day of project.

To predict the performance of the air cooled chiller it is required i) cooling loads profiles of the building and ii) database of the dry bulb temperature (DBT) at the region the system is being planned. This information should be obtained during an appropriate period and they should be used to simulate the system.

To demonstrate the method it is used a one day profile of cooling load and DBT. Table 1 shows the dry bulb temperature and cooling load during a day of project (Piatto, 2007). The cooling load profile belongs to a commercial building, since the cooling load follows to night period, different of a pattern profile that is more intense during the day and mild at night.

According to the cooling load profile the daily refrigeration consumption of the system is 6733 kWh (1915 Ton.h) and the system is assumed to operate between 10:00 to 22:00 hours, period of occupancy of the building.

The highest cooling load of the building is used to determine the chiller nominal capacity. A security factor is used since the chiller can achieve a lower capacity at hours of high DBT. In the case study, the highest cooling load is 633 kW (180 TR) and it will be assumed a security factor close to 10%, giving a chiller capacity of 700 kW (200 TR). It was selected a model of an evaporator, condensers and compressors that satisfies this requirements. The chiller arrangement

Table 1. **Dry bulb temperature and cooling load of the building in study.**

Hour	DBT ($^\circ$C)	Cooling demand (kW)
01:00	18	0
02:00	18	0
03:00	17	0
04:00	16	0
05:00	15	0
06:00	16	0
07:00	18	0
08:00	19	0
09:00	20	0
10:00	24	334.02
11:00	30	439.5
12:00	32	527.4
13:00	32	597.72
14:00	33	615.3
15:00	32	632.88
16:00	30	615.3
17:00	28	597.72
18:00	26	527.4
19:00	25	492.24
20:00	24	474.66
21:00	23	457.08
22:00	21	421.92
23:00	19	0
00:00	19	0

that was chosen is shown in Fig. 1 and has the following characteristics:
- Nominal capacity of 200 Tons with condensation by air;
- Two refrigeration circuits with 3 compressors each one, the compressors are fabricated by BITZER.
- Both circuit, circuit 1 and circuit 2 utilize one finned coil type condenser and six 1.5 kW fans.
- One common evaporator for the two circuits. The evaporator is a shell-tube heat exchanger.

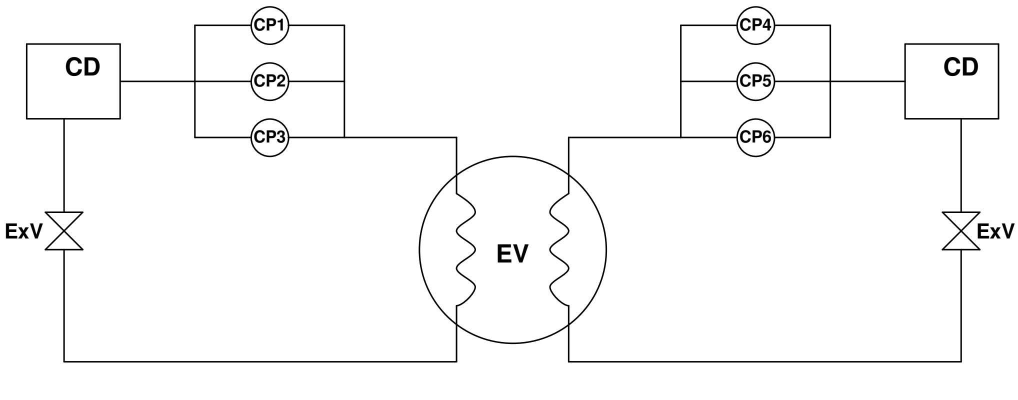

Figure 1. **Arrangement of the vapor-compression chiller, the system divided in two circuits, each circuit with 3 compressors 6G-40 (CP), one condenser (CD), one evaporator of type shell and tube heat exchanger (EV), and two expansion valves (ExV).**

Some assumptions has been made to simplify the control system of the vapor-compression system and to obtain a computational thermodynamic simulation that permit an easily and coherent operation of the equipment:
- The chiller adapts its load taking control of the number of compressors that will be turn-on; this will vary according to the cooling load, in others words, the system controls the“turn-on” or “turn-off” of additional compressors to supply the

cooling load.
- Two refrigeration circuits operating individually, however both circuit 1 and circuit 2 will supply simultaneously the cooling load.
- The six fans in each circuit will be in operating during all the time of the system operation except when no one compressor of the circuit will be turn-on, here the fans of the circuit will be turn-off.
- The flow rate of the chilled water was maintained constant during all the simulations, with a valor equal to 110.06 m^3/h.
- The evaporator leaving water temperature was established constant during all the simulations, with a valor equal to 7 °C.

3. SIMULATION RELATIONS AND HYPOTHESIS

The thermodynamic analysis was developed using the manufacturer methodology Special Mix-Match Selections. This selection procedure takes in account the data and performance curves of the equipments involved in the system (evaporator, condenser and compressors), which allow us to draw its operation parameters close to their real values.

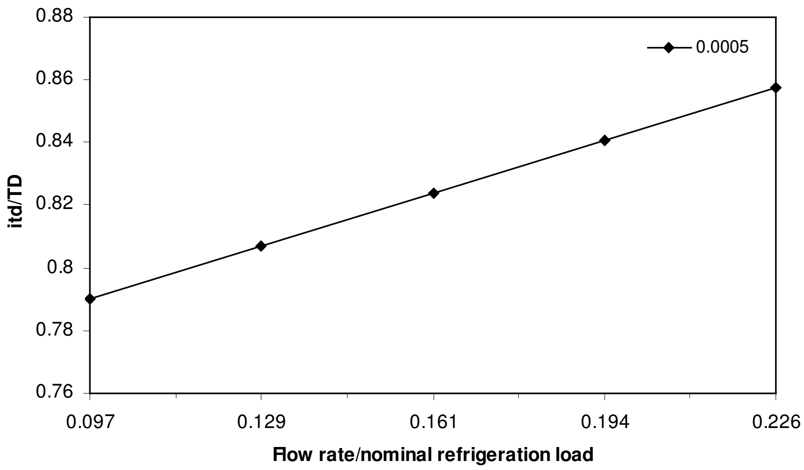

Figure 2. **Evaporator performance curve working with R-22, for a fouling factor of 0.0005. The curve as function of the ratio of chilled water flow rate and refrigeration load.**

The thermodynamic procedure begins with the determination of the chilled water flow rate for the system, which is maintened constant during all the operation time. The flow rate is calculated at the chiller nominal capacity and the design-chilled water temperature drop is 5.5 °C. The chilled flow is calculated as,

$$\dot{m} = \frac{\dot{Q}_n}{C_{p,w}\Delta T} \tag{1}$$

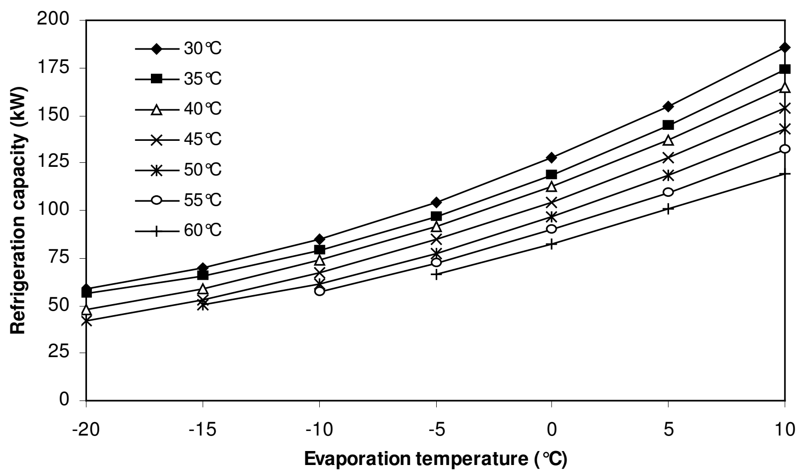

Figure 3. **Refrigeration capacity of the compressor 6G-40 working with R-22.**

The chiller is simulated considering that the chilled water always leaves the chiller at 7 °C (T_{lw}). As the cooling load is lower than the chiller capacity, the returning chilled water temperature will be lower than the design condition (12.5 °C).

An energy balance determines the chilled water temperature drop for the cooling loads during the operation time of the system, and can be written as

$$\Delta T_R = \frac{\dot{Q}_R}{\dot{m} C_{p,w}} \tag{2}$$

The ratio between the chilled water flow rate and the nominal refrigeration load is replaced in the evaporator performance curve considering a fouling factor of 0.0005, which is illustrated in Fig. 2. This procedure permits to obtain an initial temperature difference fraction (itd/TD) that will determine the evaporating temperature using the relation,

$$T_{ev} = T_{ew} - \Delta T_R.(itd/TD) \tag{3}$$

where the evaporator entering chilled water temperature is given by

$$T_{ew} = T_{lw} - \Delta T_R \tag{4}$$

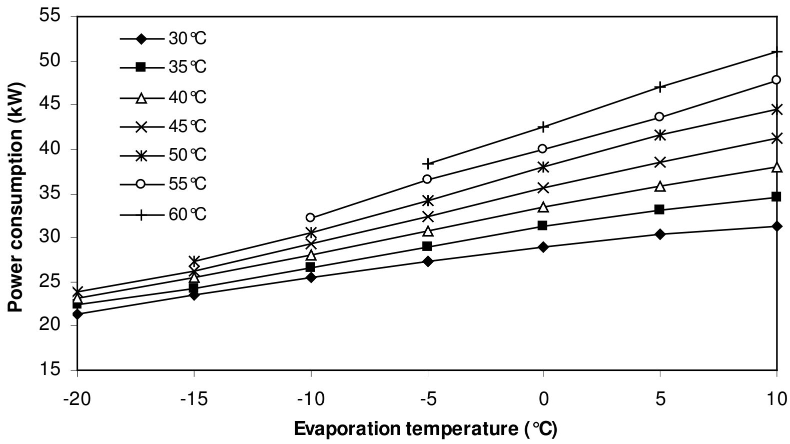

Figure 4. **Power consumption of the compressor 6G-40 working with R-22.**

At this evaporating temperature the evaporator is able to reduce the chilled water temperature from the entering condition to the leaving condition.

The refrigeration capacity and power consumption is evaluated for the calculated evaporating temperature. Since the condensing temperature is not yet defined it is utilized different condensation temperatures (T_{cd}). The compressor performance curves are shown in Fig. 3 and in Fig. 4.

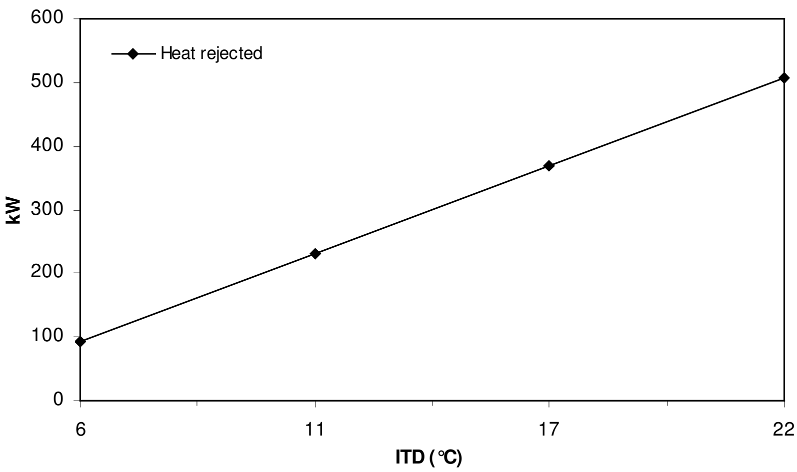

Figure 5. **Condenser performance curve. The heat rejected curve with R-22, as function of the initial temperature difference ($ITD = T_{cd} - DBT$).**

The condenser performance curve is defined as the rejected heat curve for a condenser working with the fluid refrigerant R-22 and is illustrated in the Fig. 5.

Utilizing the compressor power consumption curve (as a function of the condensation temperature and at the calculated evaporation temperature) the energy to be rejected at the condenser can be calculated for different condensing temperatures. This points are plotted in the condenser rejected heat curve to obtain the circuit balance point, this means an equilibrium point between the rejected heat and condensing temperature. The marriage curve for the demand case of 615.3 kW cooling load and $30\,^{\circ}\mathrm{C}$ dry bulb temperature is depicted in Fig. 6.

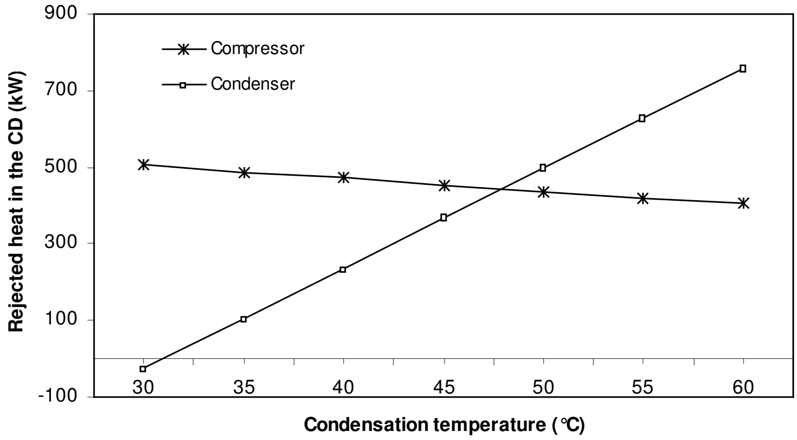

Figure 6. **Marriage curve for the compressor performance and condenser performances at a cooling load of 615.3 kW and dry bulb temperature of** $30\,^{\circ}\mathrm{C}$**.**

The balance point of the cross-plotted compressor power curve and condenser performance curve will determine the capacity of the chiller. If the chiller capacity is higher than the cooling load, compressors will be turned-off to adjust the chiller capacity to the cooling load. The number of turned-on compressors in each circuit is then determined.

The Coefficient of Performance (COP) of the circuit is determined by the refrigeration capacity and total power consumption (Moran and Shapiro, 2006). The total power includes the power consumed by the compressors and fans of the circuit. The compressor parameters (refrigeration capacity and power) are evaluated at the condensing temperature (T_{cd}) and the evaporating temperature (T_{ev}) calculated above. Finally the circuit COP may be written as,

$$COP = \frac{\dot{Q}}{\sum \dot{W}_c + \sum \dot{W}_f} \tag{5}$$

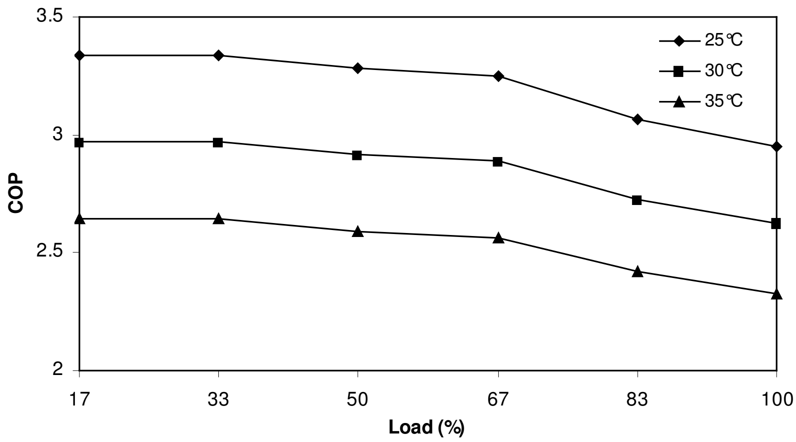

Figure 7. **Comparison of the coefficient of performance as function of the system load at different DBT.**

4. RESULTS AND DISCUSSION

4.1 Coefficient of Performance Analysis

The thermodynamic simulation procedure allows us to make an analysis of the coefficient of performance as a function of the chiller load. Three dry bulb temperatures with a constant cooling load (703 kW) were simulated to show how the coeficiente of performance (COP) decreases as the dry bulb temperature and chiller load increase (Figure 7). Higher COP's were obtained at lower temperatures (25 °C) and lower chiller loads (17%) .

Table 2. **Comparison of the coefficient of performance (COP) between two forms that the chiller.**

Load (%)	Progressive turn-off of compressors			Turn-off compressors only in one circuit		
	CPs's Circuit 1	CP's Circuit 2	COP	CP's Circuit 1	CP's Circuit 2	COP
100	3	3	2.62	3	3	2.62
83	3	2	2.73	3	2	2.73
67	2	2	2.89	3	1	2.71
50	2	1	2.92	3	0	2.62
33	1	1	2.97	2	0	2.89
17	1	0	2.97	1	0	2.97

Two diference sequence to control the capacity was investigated. In the first type (progressive turn-off compressors) one compressor is turned-off at each circuit until the chiller capacity is lower than the cooling load. In the second type (turn-off compressors only in one circuit) the compressors of one circuit are turned-off until the chiller capacity is lower than the cooling load. This control sequence and the part load COP of the chiller can be seen in Table 2.

The Table 2 illustrates the reduction of the coefficient of performance (COP) of the system when no one compressor is operating in a circuit, thus the computational program always looks for a better number of turn-on compressors in each circuit. In this study, the refrigeration system controls the compressor operation of its circuits by following the progressive turn-off compressors, that obtains better COP's when the cooling load decreases.

4.2 System Performance

The operating performance of the chiller is determined by the simulation program, which gives the number of turn-on compressors, the evaporation temperature, the condensation temperature of each circuit, the refrigeration capacity, the power consumption, the coefficient of performance for the circuits and the system total COP.

The chiller will turn-on and turn-off the compressors to attend the cooling load. The operational status of the compressors at each day hour and the refrigeration capacity of the vapor-compression system as a function of the cooling load and DBT is given in Table 3.

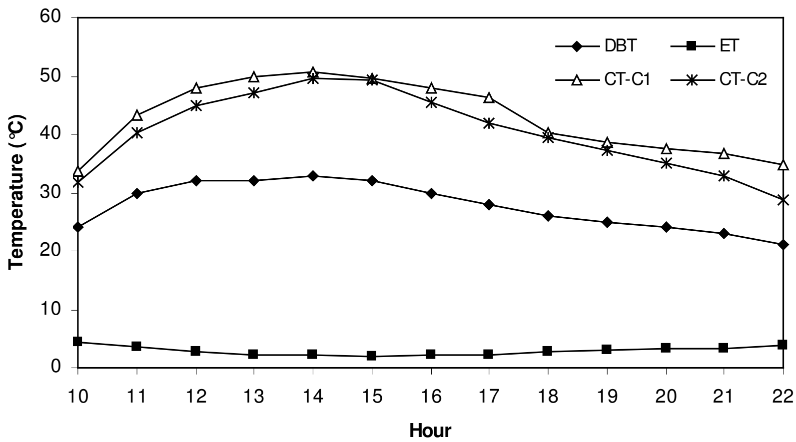

Figure 8. **Illustration of the operation temperatures during the period of demand; dry bulb temperature (DBT), evaporation temperature (ET), condensation temperature in the circuit 1 (CT-C1), condensation temperature in the circuit 2 (CT-C2).**

To adjust the chiller capacity to the cooling load, the chiller at a given hour operates between two operational modes

Table 3. **Operational configuration of the vapor-compression chiller according the period of cooling demand.**

Hour	DBT	Cooling load	Operation mode 1	Operation mode 2	Time operation mode 1	Time operation mode 2	Operation mode 1						Operation mode 2					
							CP's Circuit 1			CP's Circuit 2			CP's Circuit 1			CP's Circuit 2		
	(°C)	(kW)	(kW)	(kW)	(%)	(%)	1	2	3	4	5	6	1	2	3	4	5	6
10:00	24	334.0	421.8	295.9	30.3	69.7	X	X		X			X			X		
11:00	30	439.5	495.3	381.5	51.0	49.0	X	X		X	X		X	X		X		
12:00	32	527.4	563.6	471.8	60.6	39.4	X	X	X	X	X		X	X		X	X	
13:00	32	597.7	644.2	553.6	48.7	51.3	X	X	X	X	X	X	X	X	X	X	X	
14:00	33	615.3	632.7	543.9	80.4	19.6	X	X	X	X	X	X	X	X	X	X	X	
15:00	32	632.9	638.5	548.7	93.7	6.3	X	X	X	X	X	X	X	X	X	X	X	
16:00	30	615.3	658.5	565.7	53.4	46.6	X	X	X	X	X	X	X	X	X	X	X	
17:00	28	597.7	678.7	582.9	15.4	84.6	X	X	X	X	X	X	X	X	X	X	X	
18:00	26	527.4	608.4	508.4	19.0	81.0	X	X	X	X	X		X	X		X	X	
19:00	25	492.2	519.3	399.6	77.4	22.6	X	X		X	X		X	X		X		
20:00	24	474.7	528.0	406.2	56.2	43.8	X	X		X	X		X	X		X		
21:00	23	457.1	536.6	412.9	35.7	64.3	X	X		X	X		X	X		X		
22:00	21	421.9	426.4	298.6	96.5	3.5	X	X		X			X			X		

(number of compressors turned-on). The cooling load is compared with the chiller capacity at these operational modes and the hour fraction at each mode is deternined.

Considering hour 15:00, the cooling load is assumed as 632.9 kW. At this hour the chiller operating at full load (6 compressors in operation) can produce 638.5 kW. If one compressor of circuit 2 is turned-off the chiller can produce 548.7 kW of refrigeration. Since the chiller follows the cooling load it is assumed that it will operate 93.7% of the hour at full load and 6.3% with 5 compressors turned-on.

The operation temperatures are given in Fig. 8, where the mean condensation temperature for each circuit is determined according the percentage of time for the chosen operation modes, while the determination of the evaporating temperature is according the procedure showed above, equation (4).

The evaporation temperature has its lowest values at hours of maximum cooling load and higher dry bulb temperatures. While the results of the condensation temperature of the two circuits show that both circuit 1 and circuit 2 work independently of each other, they present a similar behavior as relation of the dry bulb temperature (higher condesation temperatures at higher dry bulb temperatures).

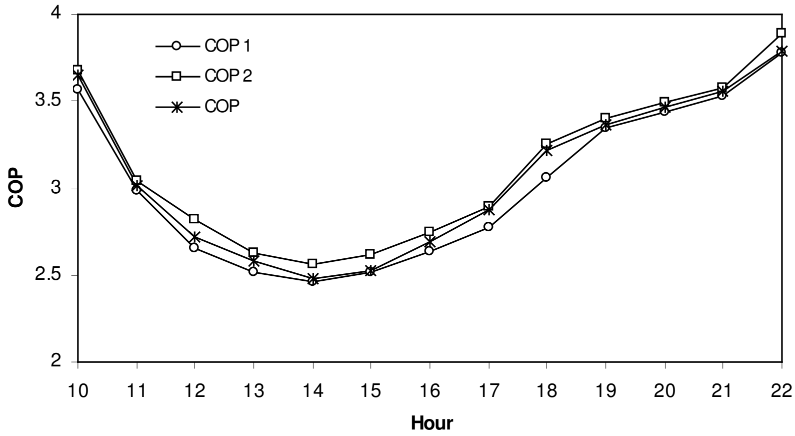

Figure 9. **Illustration of the coefficient of performance of the total system (COP) during the period of demand, the COP at operation mode 1 (COP 1), the COP at operation mode 2 (COP 2).**

The lowest evaporation temperature ($2.05\,^{\circ}\mathrm{C}$) occurs when the system has to furnish the highest cooling load (632.9 kW) and the ambient registers one of its greater temperatures of the day. In this situation, the system presents condensation temperatures equal to $49.7\,^{\circ}\mathrm{C}$ and $49.38\,^{\circ}\mathrm{C}$ in the circuit 1 and in the circuit 2, respectively. This diference between circuit condensation temperatures is due to the less one operating compressor in the circuit 2.

The coefficients of performance at operation modes (COP 1 and COP 2), and the mean COP of their combination to attend the cooling loads are given in Fig. 9. The three curves of coefficient of performance have their lowest values between hour 12:00 to 17:00, since during this period of time the system needs to supply higher cooling loads and the outside dry bulb temperature is greater (Fig. 8). According to the simulation results, the system reaches coefficients of performance between 2.48 to 3.78.

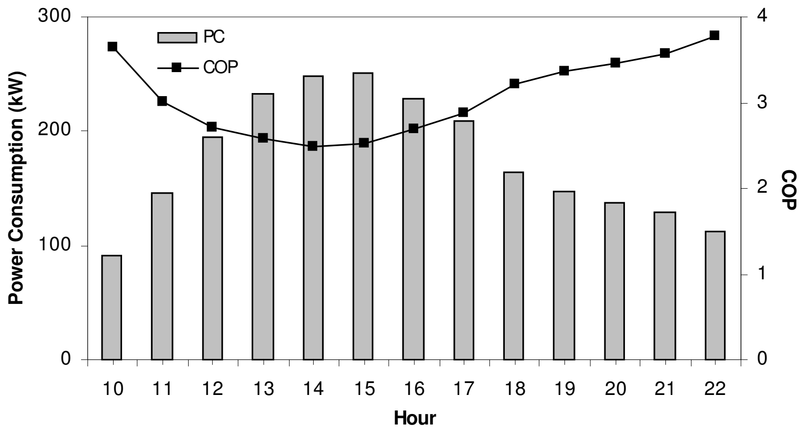

Figure 10. **Illustration of the relation between the coefficient of performance and the power consumption during the period of demand, power consumption (PC).**

The chiller power consumption and the coefficient of performance (COP) during the daily operation is given in Fig. 10. Between 12:00 to 17:00 hours due to high cooling conditions, lower evaporating temperatures and high condensation temperatures, the coefficient of performance suffer a considerably drop and high power consumptions can be seen.

The simulation methodology applied for the cooling load profile reachs an electrical energy consumption of 2290 kWh per day, included the compressors and fans consumptions. An average chiller coefficient of performance equals to 3.07 was obtained, wich corresponds to a ratio between the power consumption of the system and the cooling load during a day of operation equals to 1.196 kW per Ton.

5. CONCLUSIONS

A simulation procedure and formulations developed for the thermodynamic evaluation of a vapor-compression chiller using a project day data was utilized to predict the performence of an air cooled reciprocating compressor chiller. The results provide important information regarding the coefficient of performance of the entire system and its circuits through a mix-match selection methodology utilized by a chiller manufacturer. This identifies the energy balance point of the circuits and consequently the number of turn-on compressors.

Energy analysis of air conditioning system has become more popular in recent years and today the simulation of energy consumption in buildings appears as a good tool to get this goal. Prediction of the performance of air conditioning systems permits the comparison between different systems solutions, and the choice of the better solution based on a decision criteria (e.g. lower energy consumption, payback, etc).

The analysis of the system performance at part load can contribute in the design of the control strategy of the equipment (turn-on and turn-off the compressors). The results illustrates that is better to turn-off the operating compressors sequentially for each circuit, e.g. 2 compressors operating in each circuit is desirable than 3 in a circuit and 1 in the other.

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