



Comparative study of air conditioning systems with vapor compression chillers using the concept of green buildings

Gutenberg da Silva Pereira¹, Ana Rosa Primo¹ and Alvaro Antonio Ochoa Villa^{1,2*}

¹Universidade Federal de Pernambuco, Avenida Professor Moraes Rego, 123, 50670-901, Recife, Pernambuco, Brazil. ²Instituto Federal de Tecnologia de Pernambuco, Avenida Professor Luiz Freire, 500, 50740-540, Recife, Pernambuco, Brazil. *Author for correspondence. E-mail: ochoaalvaro@recife.ifpe.edu.br

ABSTRACT. This paper sets out to compare two different cooling systems that use vapor compression chillers for air conditioning environments. It was proposed to compare different operations in isolated and combined action operations. These operations are evaluated in the concepts of green buildings. A mathematical model was developed based on the principles of mass and energy conservation and complemented by various functions so as to determine the thermophysical properties and efficiencies of the compressors. The equations of the model were solved by the EES (Engineering Equation Solver) program. The model evaluates the influence of the main HVAC operating parameters of the chilled water system when operating under three different configurations. The results showed that the system with a differentiated compression presents a COP equal to that of the system with screw chillers in the range 0-300 RTs, and a COP that is on average 9% higher in the range 400-800 RTs.

Keywords: centrifugal chiller, screw chiller, energy, simulation, COP.

Estudo comparativo de sistemas de ar condicionado com chillers de compressão de vapor com o conceito de edifícios verdes

RESUMO. O presente trabalho se propõe a comparar dois sistemas de refrigeração diferentes usando chiller de compressão de vapor, para a climatização de ambientes. Foi proposto operações diferentes para a comparação em operações isoladas e ações combinadas. Essas operações são avaliadas nos conceitos de edifícios verdes. Foi desenvolvido um modelo matemático baseado nos princípios de conservação de massa e energia e complementado por várias funções, de modo a determinar as propriedades termofísicas e eficiências dos compressores. As equações do modelo foram resolvidas pelo programa EES (*Engineering Equation Solver*). O modelo avalia a influência dos parâmetros operacionais principais do sistema de refrigeração de água gelada operando em três configurações diferentes. Os resultados mostraram que o sistema com compressão diferenciada apresenta COP igual ao do sistema com chillers parafuso na faixa de 0 a 300 TRs, e um COP superior em média de 9% na faixa de 400 a 800 TRs.

Palavras-chave: chiller centrífugo, chiller de parafuso, energia, simulação, COP.

Introduction

The number of buildings has grown dramatically in recent decades, thus increasing the demand for air conditioning. It becomes imperative to seek optimum design, installation, maintenance and operation (WARREN; TAYLOR, 2008; YU; CHAN, 2008; CARVALHO et al., 2012; SCHIBUOLA; SCARPA, 2014; OCHOA et al., 2015), so that the system consumes less energy, thereby air conditioning within the technical standards.

The concept of a Green building, also called a sustainable building, is about creating structures and using processes that are environmentally responsible and resource-efficient throughout a building's

life-cycle, in an optimal way (JUDELSON, 2007; MA; WANG, 2009; SETYOWATI et al., 2013). So that buildings are considered green, they must follow strict precepts and determinations such as meeting national and international green standards for: construction, air quality; energy use; water use; labor safety and hygiene of the work environment; use of environmentally friendly materials; ergonomic fixtures and fittings; correct treatment of solid waste and controlling emissions. As for energy efficiency, green buildings should seek strong alternative energy sources and have emergency arrangements to ensure lighting in case of accidents; control consumption and pursue total efficiency (KASAI; JABBOUR, 2014; WEI et al., 2015; EPA,

2015; HASSOUNEH et al., 2015). The Green buildings concept also prompted the creation of LEED (Leadership in Energy and Environmental Design) certification, which is the most best-known certification used for evaluation protocols and environmental certification in the world (PAUMGARTTEN, 2003).

According to the Green Building Council Brazil, constructing new buildings in accordance with the concept of green buildings can provide customers with several advantages such as a greater appreciation in value of the property, the resale value of which can increase by 15%. In addition, using this concept can decrease waste by approximately 80% and also reduce the consumption of water and electricity by 50 and 30%, respectively (GBC, 2012).

Various studies have shown that an air conditioning system is one of the largest consumers of energy depending on the external environmental conditions (LOMBARD et al., 2011; ELAHEE, 2014; WALKER et al., 2014; QI; LU, 2014; VAKILOROAYA et al., 2014b; HITCHIN et al., 2015). Depending on the geographical region in which a building is located, an air conditioning system can consume from 40 up to 60% of all electricity used in the building (POORIWAT; PIPAT, 2011; YIM, 2014). Hence the importance of using more efficient and optimized systems that enable power consumption to be as low as possible. However, this still has to be with a cooling capacity that adequately produces thermal comfort. What this entails has been set out by various authors (MUI et al., 2010; RAYMOND et al., 2011; BILAL et al., 2013; QURESHI et al., 2013; HASSAN, 2013; JAIN et al., 2014; FARAJI et al., 2014). Various configurations of air conditioning can be used depending on the local environmental parameters involved (VAKILOROAYA et al., 2014a; SAG et al., 2015) and also, on the type of water pumping system, the thermal load, the compression system, the condensed water cooling system and control systems (QI; DENG, 2009).

Different authors have analyzed the First and Second Laws of Thermodynamics to measure the performance of mechanical cooling systems and absorption systems and target thermal comfort (KEÇEBAS, 2013; JAIN et al., 2013; ATTAR et al., 2014; OCHOA et al., 2014a; OCHOA et al., 2014b; OCHOA et al., 2014c; LIN et al., 2014; AMINYAVARI et al., 2014; JAIN; ALLEYNE, 2015).

Milazzo and Gianetti (2014) conducted a thermodynamic analysis on a regenerative cooling

system with air condensation with a view to suggesting improvements in the configuration and potential of the cooling cycle. Similarly, this analysis can be applied to compare the behavior of different cooling fluids and thereby to seek the best performance in terms of the cooling capacity of the equipment (NAPHON; WIRIYASART, 2012; APREA et al., 2013; ZOU; HRNJAK, 2014; SHE et al., 2014).

Very often, the key to reducing costs for green construction is associated with the cost and performance interrelationships that exist between the different systems (WARREN; TAYLOR, 2008; YU et al., 2015), as has been discussed in different papers in the pertinent literature that target green buildings (MA; WANG, 2009; YAN et al., 2010; DENG et al., 2011; BYRDA; LEARDINIA, 2011; DALL'O et al., 2012; HEPBASLI, 2012). However, the range of application with respect to both condensing and evaporating temperatures should be taken into account, but so too should the thermal load demand and the type of operation required by the air conditioning system.

This paper sets out to conduct a comparative study of two compression systems in chillers, with isolated and combined operation, that air-condition environments for the concept of green buildings, by developing a theoretical-experimental model based on the principles of conserving energy and mass.

Material and methods

The methodology used can be described as:

- Methodology for the experimental study;
- Methodology for the Numerical Simulation.

In the methodology for the experimental study, a description will be given of the systems to be analyzed with regard to the refrigerant, chilled water and condensate in addition to which an experimental analysis of an actual installation is undertaken.

In the methodology for the numerical simulation, energy modeling of the refrigeration and hydraulic components of the system and a global analysis of the system are undertaken. In the case study, a comparison is made between the actual data of the installation within a range of applications and numerically stimulated data from the same system. Additionally, several numerical simulations were performed with a view to identifying the range of applications and the best configuration of the chillers, in which the performance of the system would be improved.

Description of a real chilled water system

For the real case study, an analysis was made of the chilled water system used in the 11-storey building of the Ministry of Finance (Regional Bodies) in the South Sector of Local Authorities in Brasilia, which had new chillers and water pumps, one of the chillers being of the centrifugal type and another of the screw type, both of which have 400 RT cooling, as shown in Figure 1.

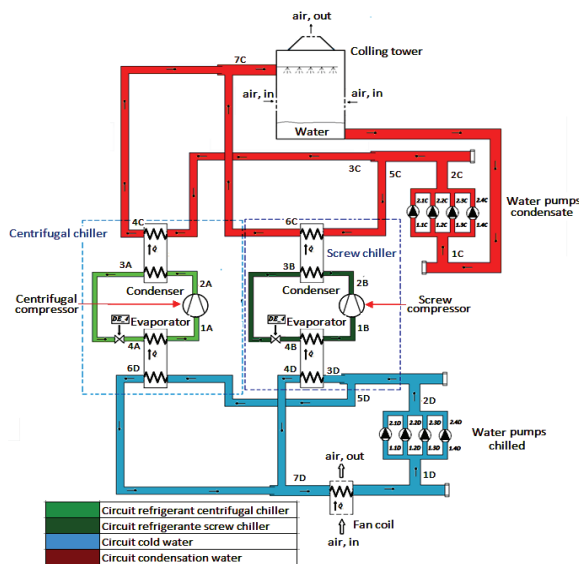


Figure 1. Chilled Water System.

The system can be divided into four circuits, the circuits being of centrifugal and screw cooling, with similar functions, but differences in the type of compression used. These cooling circuits work with R-134a refrigerant.

The third circuit is that of water condensate, which is circulated through the shell and tube exchanger (condenser side), where heat is exchanged with the refrigerant, and this heat is rejected in the cooling tower. The fourth circuit is that of chilled water, which is circulated through the shell and tube exchanger (evaporator side), where it exchanges heat with the refrigerant, thus cooling the water which will go to the fan coils.

This system has 4 water condensate pumps (Manufacturer: IMBIL, model: INI 50-160), there being 1 reserve pump and 4 pumps of chilled water (Manufacturer: IMBIL, model: INI 50-125), with 1 reserve pump.

The system also has 8 cooling towers (Manufacturer: Alpina 770,000 kcal hour⁻¹ each), 44 fan coils (Manufacturer: Carrier, model 40RR010) and 2 chillers with different compression, one of which is of the centrifugal type

(Manufacturer: Carrier, Model: 19 XRV 400 RTs) and the other of the screw type (Manufacturer: Carrier, Model: 23 XRV 400 RTs).

As to the operation of the centrifugal and screw chillers what will be used are the best ranges of application with regard to the performance of each, as per what the thermal load requires. The Measurements were taken for 30 consecutive days as follows:

- Flow rate of the chilled water and the water condensate;
- Inlet and outlet pressures and temperatures of the chilled water and also condensation temperatures of the shell and tube exchangers and, in the refrigerant lines the temperatures of discharge and suction of the compressor and also of the liquid line of the chillers;
- Power consumption of the chillers.

Numerical modeling of the components of the chilled water system

To model the energy of the systems, the First Law of Thermodynamics was applied to each separate component of the system, as per Equation 1, considering the following simplifying assumptions (BILAL et al., 2013; QURESHI et al., 2013; OCHOA et al., 2014a; OCHOA et al., 2014b): that the process is on a permanent basis; the kinetic energy variations are negligible between the input and output of all components of the cycle; potential energy variations are negligible between the input and output of all components of the cycle; the expansion and compressor device are adiabatic and the pressure drop and heat transfer in interconnection pipes are disregarded.

$$\frac{\partial E_{cv}}{\partial t} = \dot{Q}_{cv} - \dot{W}_{cc} + \sum_{in} \dot{m}_{in} \left(h_{in} + \frac{v_{in}^2}{2} + gz_{in} \right) - \sum_{out} \dot{m}_{out} \left(h_{out} + \frac{v_{out}^2}{2} + gz_{out} \right) \quad (1)$$

The nomenclature for the energetic modeling is express as:

The energy flows represent the heat flow (\dot{Q}) and power (\dot{W}), (\dot{m}) represents the mass flow, (h) is the specific enthalpy, (ϑ) and (V) are the specific and absolute volume, (c) and (C) are the specific heat capacity and heat capacity, (η) and (ε) are the efficiency and effectiveness, (ρ) is the density, (I) and (U) are the electric current and the voltage. For the subscript, (*int*) and (*out*) representing inlet and outlet stream, (*evap*, *cond*, *comp*, *ED*) representing the main devices from the systems as evaporator, condenser, compressor and expansion

valve, the letters (*ele, ref, rep, rev, ef, is*) representing the electricity, the refrigerant, the makeup water, reversible, the effective and isentropic, and finally the letter (*v*) represented the space volumetric.

Modeling of the components of the central chilled water system

First, each of the system components was modeled individually and then combined to describe the performance of the entire chilled water system. The EES platform was used.

For an adiabatic compressor, there is (equation 2):

$$\dot{W}_{comp} = \frac{(h_{comp,in} - h_{comp,out})\dot{m}_{comp}}{\eta_{ele}} \quad (2)$$

The isentropic efficiency is given by Equations 3 and 4, as:

$$\eta_{is} = \frac{\text{Isentropic Compressor Work}}{\text{Real Compressor Work}} \quad (3)$$

$$\eta_{is} = \frac{\dot{m}_{comp}\Delta h_{is}}{\text{Real Compressor Work}} \quad (4)$$

As to the electrical efficiency, with the current and voltage of each 3-phase compressor, this is expressed by Equation 5 as:

$$P_{comp} = I\sqrt{3} U \eta_{ele} \cos\varphi \quad (5)$$

Equations 6 and 7 were used to determine the heat transfer in heat exchangers (INCROPERA et al., 2007), wherein cp_{water} is the specific heat of water.

$$\dot{Q}_{hx} = \dot{m}_{hx,water} cp_{water} (T_{hx,water,out} - T_{hx,water,in}) \quad (6)$$

$$\dot{Q}_{hx} = \dot{m}_{hx,ref} (h_{hx,ref,out} - h_{hx,ref,in}) \quad (7)$$

For the expansion device, it is used the Equation 8 is used:

$$h_{val,in} = h_{val,out} \quad (8)$$

Assuming an incompressible and isothermal pumping process for the water pumps, Equations 9 and 10 express the energy of the pump as follows:

$$\dot{W}_{pump} = \frac{\vartheta \cdot \dot{m}_{pump}}{\eta_{pump}} (\Delta P_{pump}) = \dot{m}_{pump} (h_{pump,in} - h_{pump,out}) \quad (9)$$

$$\dot{m}_{pump} = \dot{V}_{pump} \cdot \rho \quad (10)$$

Where ΔP_{pump} is the difference of pressure between the outlet and the inlet of the recirculation pump, ϑ is the specific volume of the fluid to be

pumped into the pump inlet and η_{pump} is the total efficiency of the pump.

For cooling towers, considering the heat and mass transfer only in the direction normal to water and air flow (the transfer to the environment is negligible) and that the specific heat of air and water are constant for a given operational situation, and it is considered that the tower cross-sectional area is uniform, that the temperature along the water flow in any normal to the flow cross section does not vary and that the heat transfer from the tower fans, air is negligible, therefore, Equation 11 can be used.

$$h_{air,tower,out} = h_{air,tower,in} + C p_{water} (L/G) (T_{water,tower,in} - T_{water,tower,out}) \quad (11)$$

Where (L/G) represents the fraction of mass flows of water and air in the cooling tower

The isentropic efficiency is considered to be constant, hence the enthalpy of the fan output is given by Equations 12 and 13 below:

$$h_{air,out} = \left(\frac{h_{air,out,rev} - h_{air,in}}{\eta_{fan}} \right) + h_{air,in} \quad (12)$$

$$\dot{W}_{fan} = \dot{m}_{air} (h_{air,in} - h_{air,out}) \quad (13)$$

Where $h_{air,out,rev}$ is the enthalpy of the air outlet if the fan is operated reversibly. When the output enthalpy is known, the energy of the fan is determined as per Equation 13.

The housing ventilation comprises a fan, a cold coil (Fancoil) and dampers, which are used to adjust the air, as shown in Figure 2.

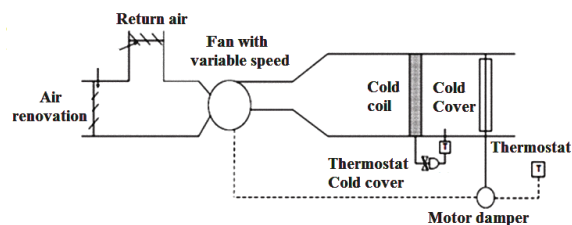


Figure 2. Air Mixing Box. (Ventilation Box) (TIRMIZI et al., 2012).

The ventilation housing can also be equipped with a variable speed fan to vary the volume of air depending on the thermal load in order to maintain an environment with thermal comfort conditions at a constant temperature of 24°C and a maximum relative humidity of 55%. A constant ratio of 15 up air and 85% return air is maintained in the air mixing box (ventilation housing). The air mixer is modeled using the effectiveness-number of transfer units (NTU) method, while the air treatment load is

assumed to be a dry bulb temperature function. Equations 14 to 17 are used to describe the air mixing.

$$\text{box. } \dot{Q}_{\text{air,box}} = \dot{m}_{\text{water}} c_{p\text{water}} (T_{\text{water,out}} - T_{\text{water,in}}) \quad (14)$$

$$\dot{Q}_{\text{air,box}} = \dot{m}_{\text{water}} c_{p\text{ef}} (T_{\text{air,in}} - T_{\text{air,out}}) \quad (15)$$

$$\dot{Q}_{\text{air,box}} = \varepsilon_{\text{min}} C_{\text{min}} (T_{\text{air,in}} - T_{\text{water,in}}) \quad (16)$$

$$c_{p\text{ef}} = \frac{h_{\text{air,in}} - h_{\text{air,out}}}{T_{\text{air,in}} - T_{\text{air,out}}} \quad (17)$$

Where $c_{p\text{ef}}$ is the effective heat capacity of the air mixing box.

Results and discussion

To validate the computational model for the centrifugal chiller, the manufacturer's data and data collected from the real system were used and then a simulation was made where, as input data were varied, the cooling capacity of the 400 RT chillers in steps of 25, 50, 75 and 100% and the evaporation temperature of the shell and tube evaporator were kept at 3°C and the condensation temperature at 39°C.

In Figure 3 shows that the simulated values and those of the manufacturer of the centrifugal chiller have a low error rate, namely 0.02% and the simulated values compared to the actual system have a permissible error of 8%.

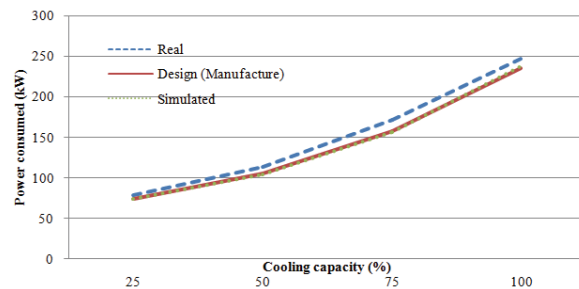


Figure 3. Consumed Power of the 400 RT Centrifugal Chiller.

In Figure 4 shows that the simulated values and those of the manufacturer of the screw chiller have a low error rate, namely 0.03% and the simulated values compared to those of the actual system have a permissible error of 7%.

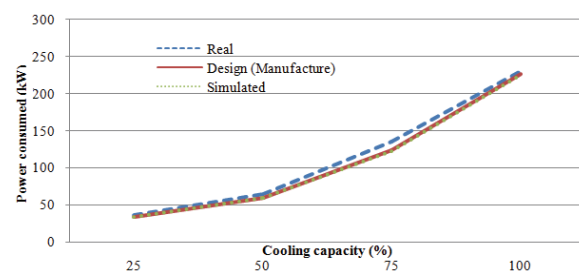


Figure 4. Consumed Power of the 400 RT screw Chiller.

COP of the chillers

On varying the capacity of the compressors and using the input data from the Table 1, it note that the performance of the chiller with a screw compressor at most thermal loads is better than that of the one with centrifugal compressor but at 90-100% of the load required, it is the performance of the chiller with the centrifugal compressor is 9% better (see Figure 5).

Table 1. Simulation Data.

Cooling Capacity	100%	75%	50%	25%
Chilled Water flow	60 l s ⁻¹	61 l s ⁻¹	62 l s ⁻¹	63 l s ⁻¹
Cooling Water flow	75.7 l s ⁻¹	75.7 l s ⁻¹	75.7 l s ⁻¹	75.7 l s ⁻¹
Condensation Temperature	36°C	36°C	36°C	36°C
Vaporization Temperature	2°C	2°C	2°C	2°C
Outlet chilled water temperature	6°C	6°C	6°C	6°C
Inlet chilled water temperature	12.2°C	10.8°C	9.4°C	8.0°C
Refrigerant	R-134a			

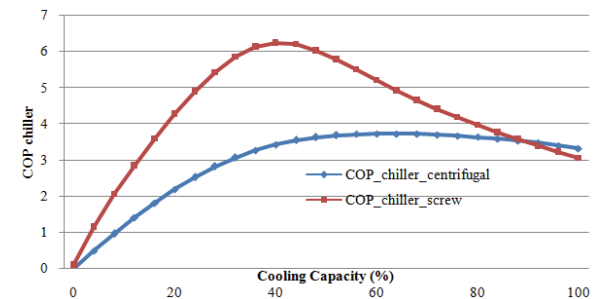


Figure 5. COP of the Chiller.

Comparison of distinct configurations for the installation

In this case, three 800 RT cooling systems with differentiated compression were simulated, for which the results are shown in Figure 6, there being:

- COP_{mix} : two 400 RT chillers were used, one of which had a screw compressor (used alone for loads from 0 to 37.5% corresponding to 0 to 300 RTs). And the other chiller with a centrifugal compressor was used alone for 43-50% of the installation, corresponding to 350 to 400 RTs. It was also considered. The two chillers working together for 56-100% loads, corresponding to 450-800 RTs, was also considered. In this case, both the centrifugal and the screw chiller at full load can vary their cooling capacity as needed.

- $COP_{\text{centrifugal}}$: two centrifugal chillers were used, where one of the chillers will work alone from 0 to 50% capacity, and from there will always remain at full load and thereafter, the second chiller is actioned at partial loads.

- Note: A rotating system of compressors that will work at full load should be adopted, thus ensuring a longer useful life for the chillers.

• COP_{screw} : in this, two screw chillers were used, where one of the chillers will work alone from 0 to 50% capacity, and from there will always remain at full load and after 50% of this, the second chiller is actioned at partial loads.

Note: A rotating system of compressors that will work at full load should be adopted, thus ensuring a longer useful life for the chillers.

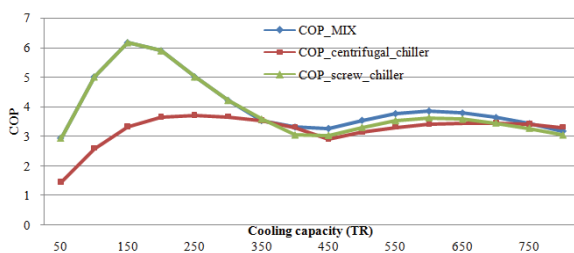


Figure 6. COP Comparison of different configurations of the operation of the chillers.

Conclusion

On varying the capacity of the three different configurations for the system, we observe that the system with two different chillers (COP_{mix}) has an equal or better performance in all the cooling capacity ranges required, with a COP that is 37% higher than that for the system with two centrifugal compressors for the range 0 to 300 RTs and a 10% higher in the range 450-750 RTs.

The system with different compression presents a COP equal to that of the system with screw chillers in the range 0-300 RTs, and its value is 9% higher in the range 400-800 RTs.

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