SIMULATION USED FOR THE SELECTION AND SIZING OF A CENTRALIZED COOLING PLANT : METHODOLOGY AND LIMITATIONS

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Topic

Nr 9 : Modelling and Experimental Approaches

ABSTRACT

During the last ten years, many HVAC simulation models and calculation tools were developed. However, until now, the practical use of simulation tools is still very limited in building and HVAC design as well as in the development of control strategies.

This paper focuses on the selection and sizing of a centralized cooling plant. Its purpose is to show how the simulation can help the designer and to point out the difficulties encountered by the simulation user.

In the design stage, various alternative cooling systems are available. Since the paper stresses on the methodology, only two design alternatives of centralized chilling plant are considered: with air-cooled reciprocating chillers and with water-cooled twin-screw chillers. The sizing of these two cooling plants is based on the sizing criteria and the building thermal loads in design conditions.

The models used to simulate the primary equipment are extracted from the simulation toolkits developed by ASHRAE and are briefly described in the paper. The model parameters need to be adjusted to represent accurately the behavior of the components. At the design stage, the information available about the components is provided by the manufactures' catalogs. The paper describes the data usually supplied by these catalogs and analyses the difficulties encountered during the procedure of parameter identification of the models.

The simulation of the thermal behavior of both cooling plants is handled by the modular simulation program TRNSYS. It is based on the identified component parameters, but also on the implementation of regulation rules.

The simulation allows the designer to check if both cooling plant alternatives can meet the building cooling loads and to estimate the energy consumption and, thus, the operating cost associated with each design alternative. The paper also points out that the auxiliary consumption's (fans, pumps) have to be considered in the determination of the operating cost.
In addition to the comparison of design alternatives, the simulation can be used at the design stage to evaluate the effects of variations of the characteristics of the systems considered. The paper shows how the system coefficient of performance is affected by an increase of the set point water temperature at the exhaust evaporator and by a different sizing of the water-cooled twin-screw chillers.

1. INTRODUCTION

The air-conditioning system selection and design depend on several interrelated criteria such as performance requirements, spatial requirements, first cost, operating cost, maintainability, etc. (ASHRAE, 1996).

Although the whole HVAC design can not be handled by the simulation tools, the simulation can help the designer by facilitating the comparison of the thermal performance of several design alternatives.

The purpose of this paper is to show how the simulation tools can deal with the data currently available during the HVAC design and to point out the difficulties encountered by the simulation user. These difficulties can appear during the procedure of parameter identification of the component models or during the comparison of the design alternatives.

When developing an air-conditioning system, numerous options are available to the designer. For example, when designing a centralized cooling plant for a large systems, several types of refrigeration equipment can be considered : reciprocating compressors, twin-screw compressors, centrifugal compressors and absorption chillers (ASHRAE, 1996). Thermal storage systems as well as several types of condensers (air-cooled condensers, evaporative condensers, water-cooled condensers associated with cooling towers) can also be taken into account by the designer.
Since this paper focuses on the methodology, only two alternatives of centralized cooling plant are considered : the first one is characterized by air-cooled reciprocating chillers and the second alternative is characterized by water-cooled twin-screw chillers associated with counterflow cooling towers.

2. DESCRIPTION OF BOTH ALTERNATIVES OF COOLING PLANT

In the first design alternative, the building cooling loads are met by two air-cooled chillers in parallel arrangement (see Figure 1). Both chillers are assumed to be identical and are equipped with reciprocating compressors.

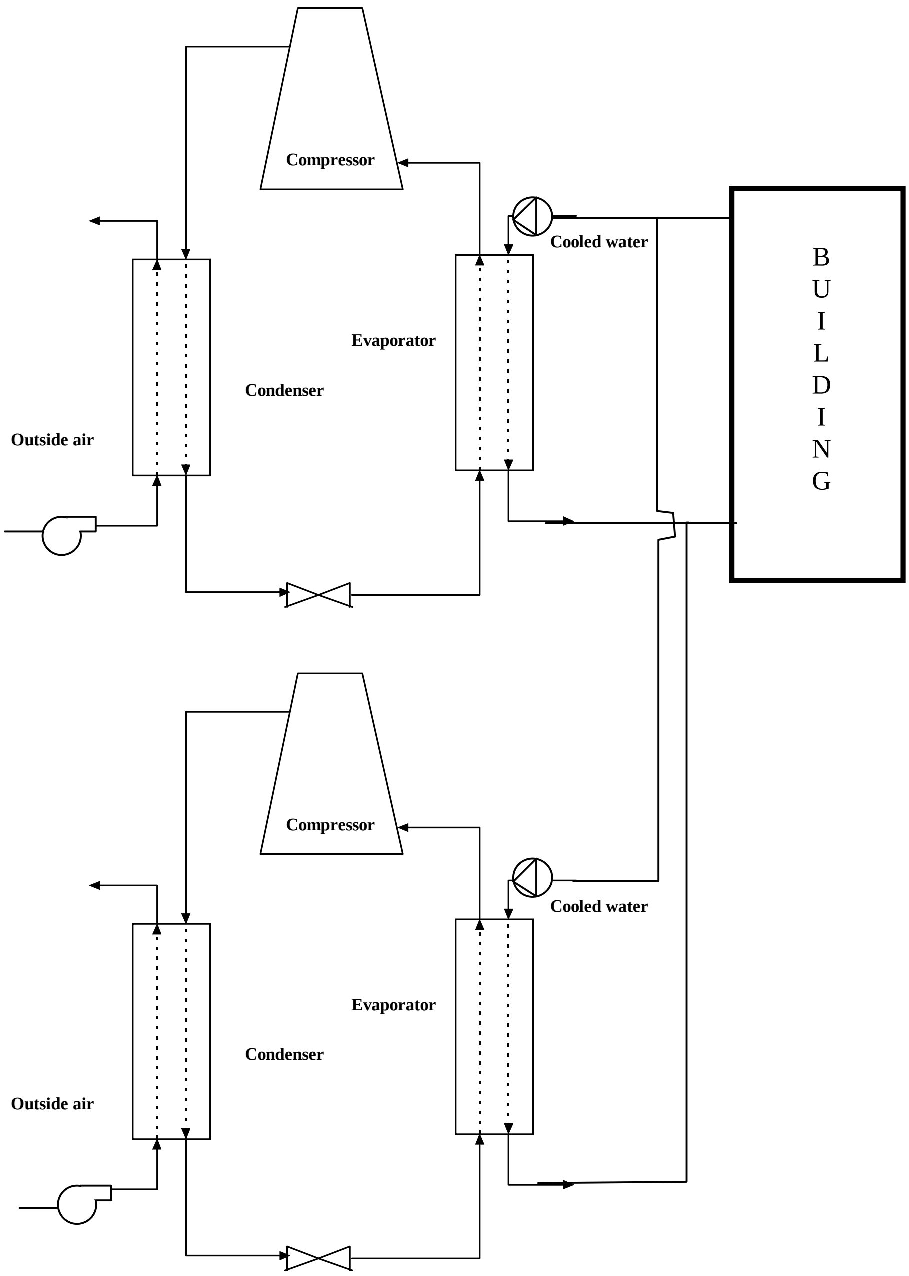

Figure 1 : Cooling plant with air-cooled chillers

In the second design alternative, the building cooling loads are met by two water-cooled chillers in parallel arrangement (see Figure 2). Both chillers are assumed to be identical and are equipped with twin-screw compressors.
To cool down the water leaving the condenser, three identical counterflow cooling towers are used in parallel. A bypass valve is used, at low cooling loads, to avoid a too low water temperature at the condenser supply.

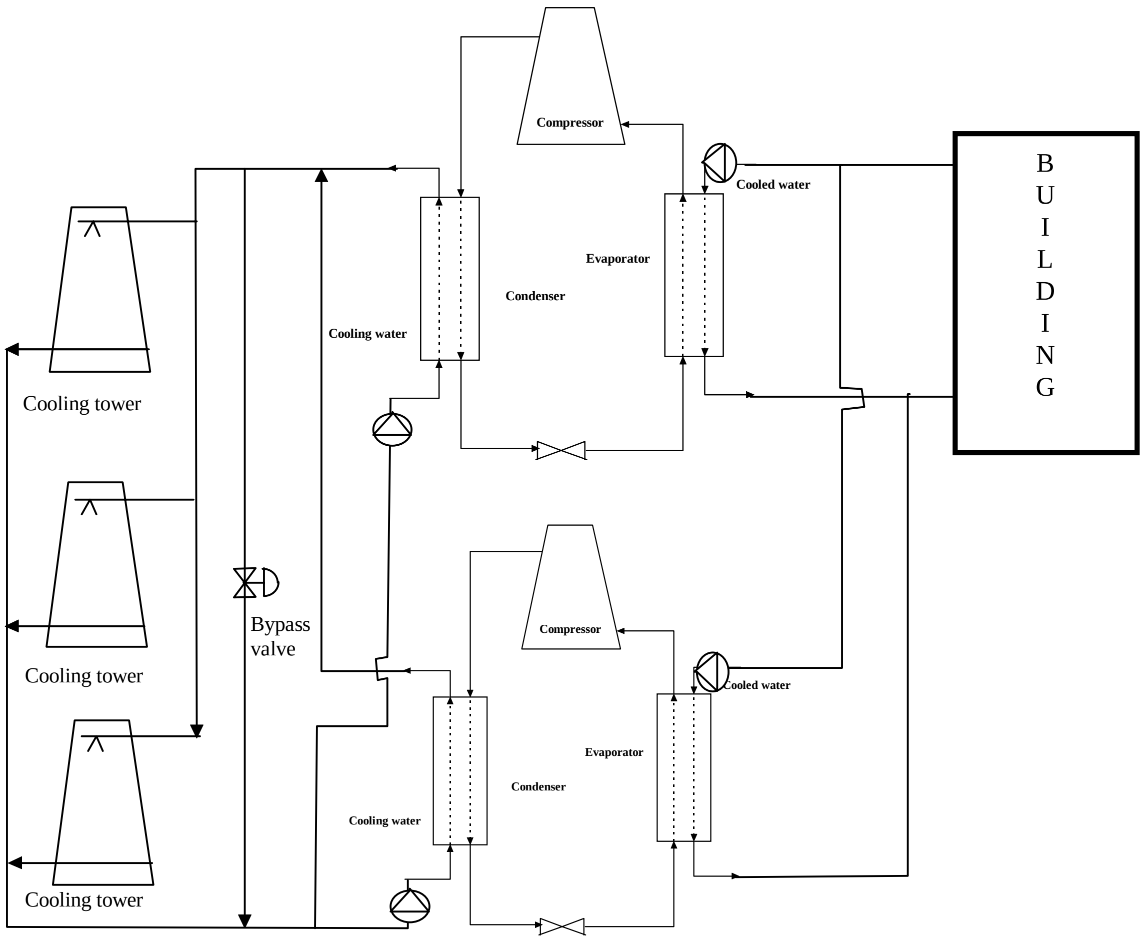

Figure 2 : cooling plant with water-cooled chillers

3. SIZING CRITERIA

During the design phase, the information available on the components consists in the data provided by the manufacturers.
To be able to select the proper component among the different models proposed by the manufacturer, the designer must consider sizing criteria. The sizing criteria define the design operating conditions of the cooling plant.

For the chillers, these conditions are :

- the design cooling loads. The cooling plant must be able to meet the building cooling loads in design conditions. These loads can be given by building cooling loads calculations. The contribution of the cooling of the outside air must be taken into account by these calculations. Figure 3 gives an example of the evolution of the building cooling demand on the design day where the maximum cooling loads occur;

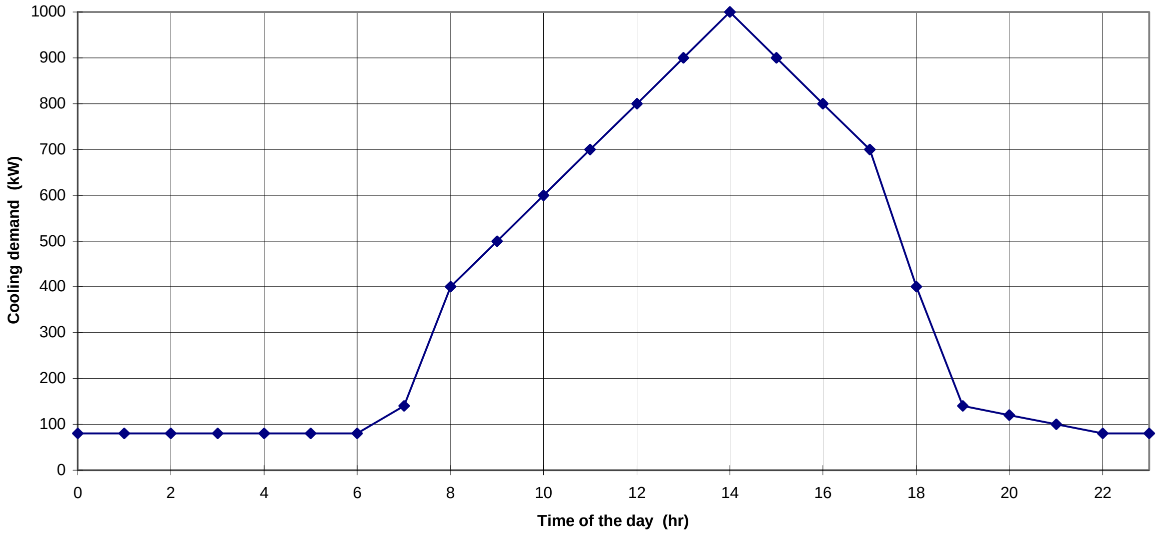

Figure 3 : Building Cooling Loads in Design Conditions

- the water temperature at the evaporator exhaust. This variable is usually a set point used for the chiller regulation. Here, this set point is assumed to be equal to 6 °C and is the same for both chillers ;
- the water flow rate in the evaporator. It is usually defined so that the water temperature drop in the evaporator is about 5 K in design conditions (maximum cooling loads). Here, it is assumed that a fixed-speed pump is associated with each evaporator and provides a water flow rate equal to 24 kg s^{-1}.
- the design operating conditions at the condenser level. For the air-cooled chiller, the design condenser supply air temperature is defined (here, 35 °C). For the water-cooled chiller, the condenser water flow rate and condenser supply water temperature are defined in design conditions. Here, the condenser supply water temperature is equal to 29 °C and, such as for the evaporator, a fixed-speed pump is assumed to be associated with each condenser and is assumed to provide a flow rate of 31 kg s^{-1} (this flow rate corresponds to a water temperature drop in the condenser of about 5 K in design conditions (maximum cooling loads)).

For the cooling tower, the design operating conditions are :

- the cooling tower supply wet-bulb temperature. Here, it is assumed to be equal to 22 °C ;
- the cooling tower water mass flow rate. It is deduced from the water mass flow rate in the condensers ;
- the cooling tower exhaust water temperature. It is equal to the condenser supply water temperature in design conditions (here, 29 °C) ;
- the cooling tower exhaust water temperature. It is deduced from the heat flow rate rejected from the chillers in design conditions.

On the basis of the sizing criteria and the manufacturers' catalogs, it is rather simple for the designer to select "by hand" the adequate components.

4. COMPONENT MODELING

Once the equipment is selected, the simulation tools can be used.

All the component models considered in this paper are extracted from the “Toolkit for Primary HVAC System Energy Calculation” (Bourdouxhe et al., 1995). They have been adapted to be used by TRNSYS, a modular simulation software (Klein et al., 1994).

The Toolkit models are static, simple, but physically meaningful. Thanks to the modeling approach considered in the Toolkit, a minimum number of parameters are required to reproduce the behavior of the component. These parameters can be identified on the basis of data currently available in the manufacturers’ catalogs.

The following three subsections describe briefly the modeling of the cooling tower and the reciprocating and twin-screw chillers.

4.1. Cooling tower modeling

This counterflow cooling tower model is based on Merkel's theory (Merkel, 1925). It also assumed that the moist air enthalpy can be defined by only the wet-bulb temperature and does not consider the water lost by evaporation. Thanks to these assumptions, the cooling tower can be modeled as a fictitious classical counterflow heat exchanger (see figure 4).

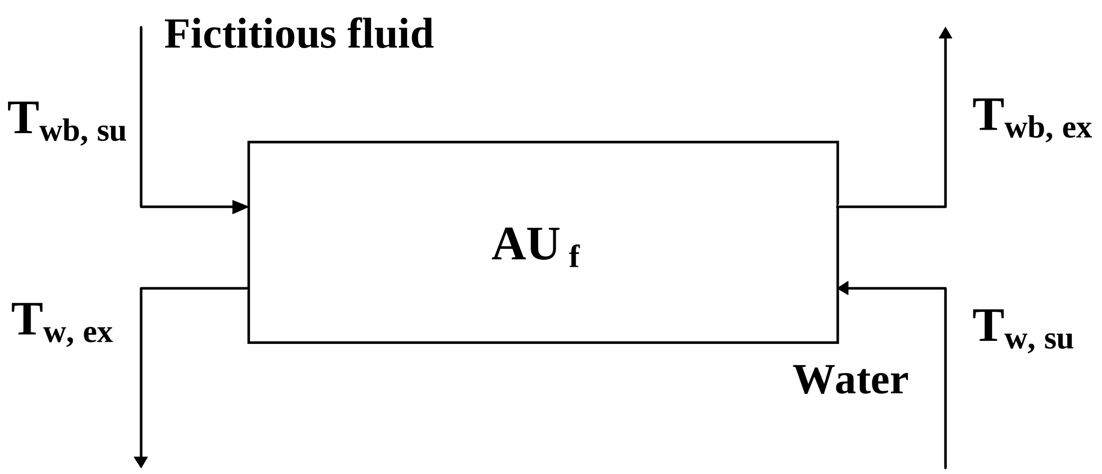

Figure 4 : Conceptual Schema of the Counterflow Cooling Tower

This heat exchanger is characterized by its fictitious global heat transfer coefficient, AU_f, which is a function of the actual global heat transfer coefficient of the tower, AU. This latter coefficient is the only parameter characterizing the tower model (at a given air flow rate).
The identification of the global heat transfer coefficient is based on available operating points obtained in steady-state regime and associated with the possible air mass flow rates. These data can be supplied by laboratory test results or by manufacturer’s catalogs.
However, it should be noted that, usually, the manufacturer’s catalogs only provide the thermal performance of the tower in highest fan speed. This means that, for cooling towers

with multiple-speed fans, the global heat transfer coefficients associated with the lower fan speeds remain usually unknown (or have to be presumed proportional to the air flow rate).

For each air mass flow rate, the parameter identification requires the following data (see Figure 5) :

- the air and water mass flow rates;
- the supply air wet-bulb temperature;
- the range (difference between the supply and exhaust water temperature);
- the approach (difference between the exhaust water temperature and the supply air wet-bulb temperature);
- the atmospheric pressure.

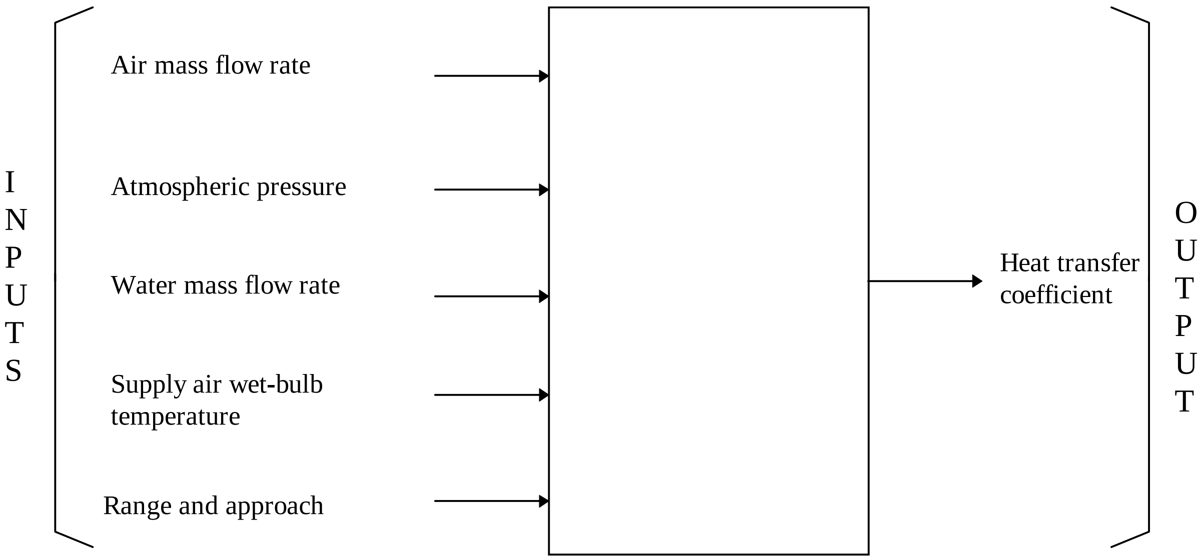

Figure 5 : Information Flow Diagram of the Cooling Tower Parameter Identification

Once the parameter(s) is(are) identified, the cooling tower can be simulated in any working conditions.

4.2. Reciprocating chiller modeling

The chiller is represented by four components : the reciprocating compressor, the evaporator, the condenser and the expansion valve (presumed to provide a perfect control) (see Figure 6). No heat exchange between this system and its environment is taken into account.

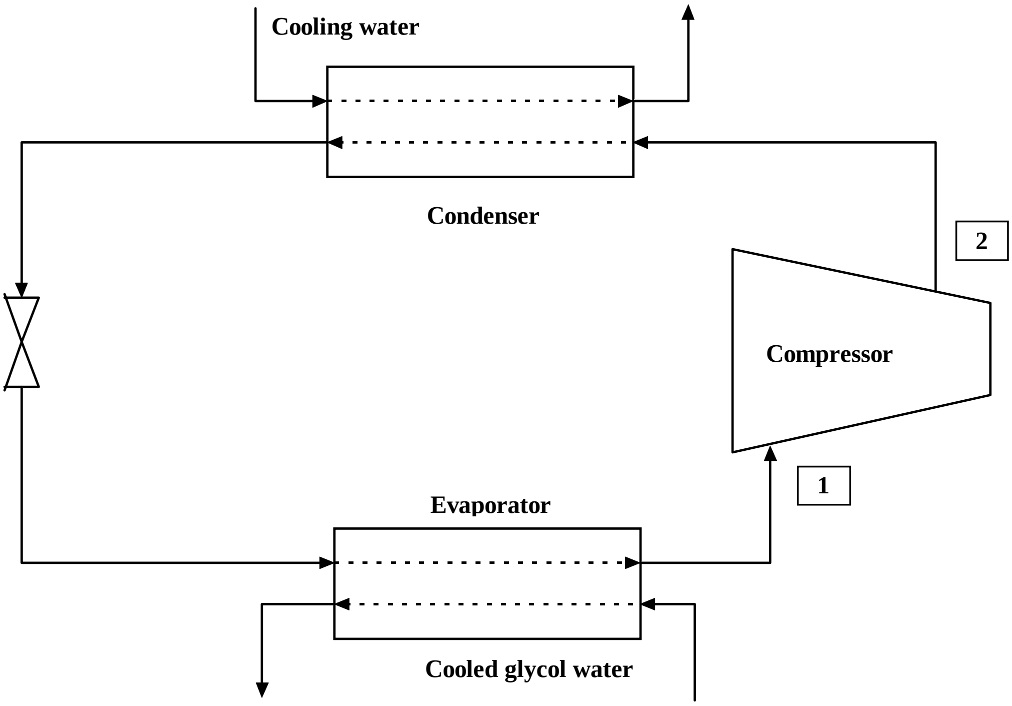

Figure 6 : Conceptual Schema of the Vapor Compression Chiller

An ideal mechanical cycle is presumed to be realized inside the compressor cylinders (Haberschill et al., 1993).

Due to the re-expansion of the clearance volume, the volume flow rate of the compressor refrigerant is a decreasing function of the pressure ratio, therefore,

$$\dot{V} = \dot{V}_{S_{FL}} \left[1 + C_f - C_f \left(\frac{p_2}{p_1} \right)^{\frac{1}{\gamma}} \right] \tag{1}$$

The parameters to be identified are $\dot{V}_{s_{FL}}$ 2 and C_f .

A simple linear model will be used in order to represent the "motor-transmission" subsystem:

$$\dot{W} = \dot{W}_{lo} + \alpha \dot{W}_{in} + \dot{W}_{in} \tag{2}$$

where $\dot{W}_{lo} + \alpha \dot{W}_{in}$ 4 represents the electromechanical losses. The new parameters, $\dot{W}_{lo}$ 5 and α also have to be identified.

The part-load operation of a reciprocating compressor is realized by cylinder unloading and by the cycling between two unloading levels or between ON and OFF regimes. The losses associated with the unloading of the compressor cylinders can be taken into account by introducing the "pumping" losses that occur in the unloaded cylinders. Due to these "pumping" losses, the internal power $(\dot{W}_{in})$ 6 is larger than the isentropic compression power. The parameter characterizing the "pumping" losses is called $(\dot{W}_{pump})$ 7 and corresponds to the

internal power of the compressor when all the cylinders are unloaded. No cycling losses are considered by the chiller model.
The refrigerant is presumed to be warmed by electromechanical and "pumping" losses before being compressed. This warming is assumed to be realized at constant pressure.

Both condenser and evaporator are represented as classical heat exchangers. No pressure drop is considered on the refrigerant side. In each of these two heat exchangers, the refrigerant side is considered to be isothermal. The condenser and evaporator heat transfer coefficients are the last two parameters to be identified in order to model the whole chiller.

Consequently, only seven parameters are needed to model the reciprocating chiller for its whole range of operation. All these parameters are presumed to remain constant.

Two parameter identifications must be performed to determine the value of the parameters. The first one deals with the parameters characterizing the full-load regime while the second one provides the parameter characterizing the part-load regime, $(\dot{W}_{pump})$.
Contrary to the cooling tower parameter identification, all the experimental data can be used when identifying the parameters of the full-load regime. Figure 7 gives the data needed for the full-load identification. As can be seen, for each working point in full-load regime, the main data needed are :

- the supply or exhaust evaporator and condenser water temperatures;
- the evaporator and condenser water mass flow rates;
- the cooling capacity and power consumed by the compressor.

On the basis of these data, the parameters associated with the chiller in full-load regime are deduced from two linear regressions and an exhaustive search method (Stoecker, 1989).

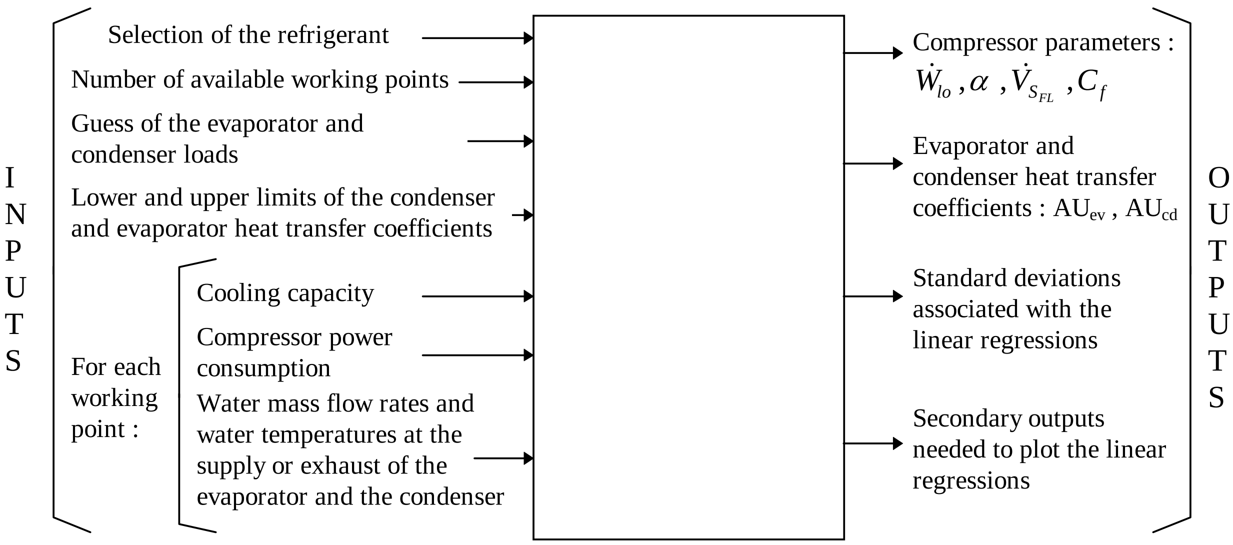

Figure 7 : Information Flow Diagram of the Reciprocating Chiller Parameter Identification (full-load regime)

The identification of the part-load parameter requires one working point in part-load regime. However, such an information is rarely supplied by the manufacturers' catalogs. Therefore, the Toolkit routine also allows, thanks to empirical relationships, the estimation of the part-load parameter on the basis of one working point in full-load regime.

Once the parameters are known, the reciprocating chiller can be simulated throughout its whole operating range. The model simulates this part-load operation by a simple linear combination of two steady-state regimes.

The Toolkit routines consider only water as secondary fluid. But it is easy to shift to other fluid. In the case of an air-cooled condenser, the air mass flow rate is replaced by an equivalent water mass flow rate. This fictitious water mass flow rate is calculated as follows :

$$\dot{M}_{w-equivalent} = \dot{M}_a \frac{c_{p_{moist\,air}}}{c_w} \tag{3}$$

4.3. Twin-screw chiller modeling

The conceptual schema of Figure 6 remains valid when dealing with a twin-screw chiller. Indeed, the twin-screw chiller modeling is identical to the reciprocating chiller modeling, except for the modeling of the compressor.

The twin-screw compressor model deals with fixed or variable-volume-ratio screw compressor. The model takes into account the two characteristics of the screw compressor : the built-in volume ratio and the internal leakage. The value of the built-in volume ratio allows one to calculate the internal compression power. Once this power is known, the compressor model uses, such as for the reciprocating chiller, the following relationship to define the motor-transmission subsystem :

$$\dot{W} = \dot{W}_{lo} + \alpha\, \dot{W}_{in} + \dot{W}_{in} \tag{2}$$

Here again, the screw compressor is also presumed to adiabatic : its losses warm up the refrigerant before compression occurs.

The "swept" volume flow rate, $\dot{V}_{s\,FL}$ 10, of the screw compressor in full-load and the internal leakage (characterized by an equivalent leakage area, A_l) allow the determination of the refrigerant mass flow rate flowing through the chiller. The parameter A_l is the only parameter that differs from the full-load parameters of the reciprocating chiller.

In part-load, moving a slide valve opens a recirculation passage and bypasses a portion of the refrigerant back to suction before much compression occurs. The part-load losses are characterized by the pressure jump, dp_{pump} , encountered by the refrigerant diverted. This is the only parameter characterizing the part-load regime. It should also be noted that the built-in volume ratio is assumed to remain constant in part-load regime.

Here again, all the parameters are presumed to remain constant during the simulation, and the heat transfer coefficients are also presumed to be independent of water flow rates.

Such as for the reciprocating chillers, two parameter identifications must be performed. The first one deals with the parameters characterizing the full-load regime while the second one provides the parameter characterizing the part-load regime.
All the experimental data can also be used when identifying the parameters of the full-load

regime. For each working point in full-load regime, the main data needed are (see Figure 9) :

- the supply or exhaust evaporator and condenser water temperatures;
- the evaporator and condenser water mass flow rates;
- the cooling capacity and power consumed by the compressor;
- the internal built-in volume ratio.

On the basis of these data, the full-load chiller parameters are deduced from two linear regressions.

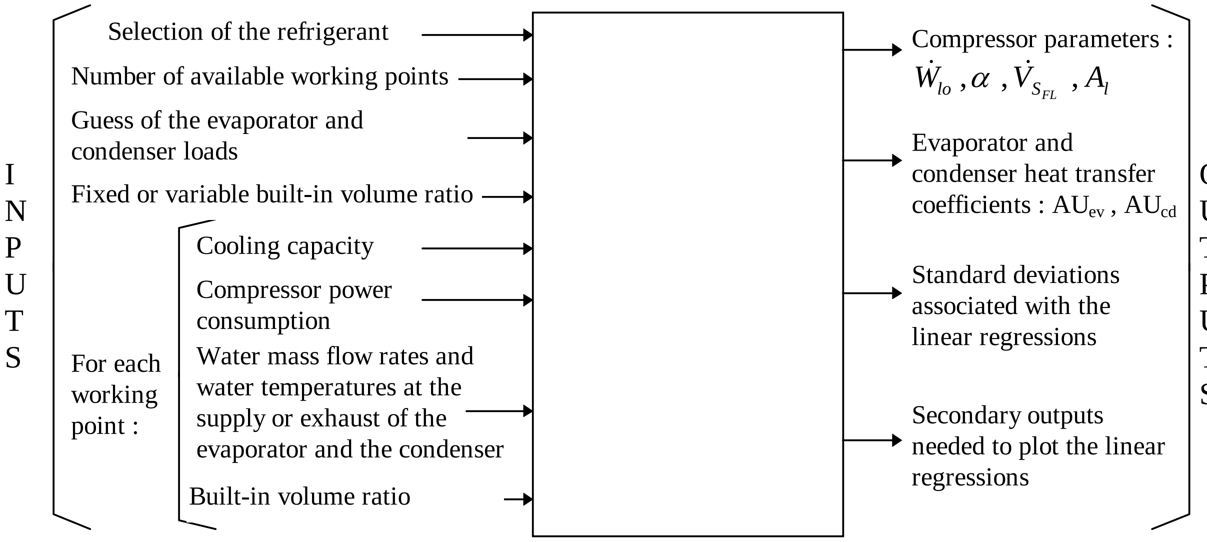

Figure 9 : Information Flow Diagram of the Twin-Screw Chiller Parameter Identification (full-load regime)

Such as for the reciprocating chiller, the identification of the part-load parameter requires one working point in part-load regime. But, here again, such an information is rarely supplied by the manufacturers' catalogs. Therefore, the Toolkit routine uses empirical relationships to estimate the part-load parameter on the basis of one working point in full-load regime.

Once the parameters are identified, the twin-screw chiller can be simulated throughout its whole operating range.

5. PARAMETER IDENTIFICATION

5.1. The water-cooled twin-screw chiller

Figure 3 and the sizing criteria specify that each chiller should be able to provide a cooling capacity of 500 kW with an evaporator water flow rate of 24 kg s^{-1} , an evaporator exhaust water temperature of 6 °C, a condenser water flow rate of 31 kg s^{-1} and a condenser supply water temperature of 29 °C.

The manufacturer 's catalog (Dunham-Bush, Unknown date)gives the thermal performance of several chiller models for a series of operating points (see Table 1). Table 1 shows that Chiller 2 can meet the design requirements. Indeed, it provides a cooling capacity of 563 kW with an evaporator water flow rate of 25 kg s^{-1} , an evaporator exhaust water temperature of

5.6 °C, a condenser water flow rate of 31.2 kg s^{-1} and a condenser supply water temperature of 29.4 °C. These flow rates are slightly higher than the design flow rate, but the influence of flow rate variations on the cooling capacity is weak (see Table 1).

Table 1 : Performance Data Provided by the Manufacturer (Water-Cooled Twin-Screw Chiller)

	Chilled water conditions		Condenser water mass flow rate and condenser supply water temperature											
			23.89 °C				29.44 °C				35.00 °C			
			24.61 kg/s		28.40 kg/s		24.61 kg/s		28.40 kg/s		24.61 kg/s		28.40 kg/s	
	Twexev	Mfrwev	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)
Chiller 1	5.56	19.69	464.64	91.00			443.52	104.00			422.40	118.00		
		21.14			478.72	92.00			457.60	105.00			436.48	119.00
	6.67	19.69	478.72	93.00			457.60	106.00			432.96	120.00		
		21.14			492.80	94.00			471.68	107.00			447.04	121.00
	7.78	20.70	489.28	94.00			475.20	107.00			450.56	121.00		
		22.15			503.36	95.00			489.28	108.00			464.64	122.00
	8.89	20.70	499.84	96.00			489.28	109.00			464.64	123.00		
		22.15			513.92	97.00			503.36	110.00			478.72	124.00
			31.23 kg/s		34.07 kg/s		31.23 kg/s		34.07 kg/s		31.23 kg/s		34.07 kg/s	
	Twexev	Mfrwev	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)	Qev (kW)	W (kW)
Chiller 2	5.56	24.99	587.84	115.00			**563.20**	**129.00**			535.04	147.00		
		26.50			598.40	116.00			573.76	130.00			545.60	148.00
	6.67	24.99	608.96	117.00			580.80	132.00			552.64	148.00		
		26.50			619.52	118.00			591.36	133.00			563.20	149.00
	7.78	26.25	619.52	119.00			598.40	133.00			570.24	150.00		
		27.76			630.08	120.00			608.96	134.00			580.80	151.00
	8.89	26.25	637.12	121.00			619.52	135.00			587.84	152.00		
		27.76			651.20	122.00			630.08	136.00			598.40	153.00

where Twexev is the evaporator exhaust water temperature (°C)
Mfrwev is the evaporator water mass flow rate (kg s^{-1})
Qev is the cooling capacity
W is the compressor power consumption

The manufacturer's catalog also provides the refrigerant used by the chiller (here, R22).

Figure 9 shows that the information still missing for the identification of the full-load parameters is the value of the built-in volume ratio. Unfortunately, this data is not always given by the manufacturer. In this case, the built-in volume ratio is assumed to be variable so that the internal pressure ratio always matches the system pressure ratio. This assumption leads to an overestimation of the electromechanical losses when applied to fixed built-in volume compressor, but the parameter identification remains satisfactory.

The identification of the full-load parameters (Bourdouxhe et al., 1995) can now be done. It gives the following full-load parameters :

$AU_{ev} = 80\,000$ W K^{-1}

$\dot{W}_{lo}$ 11 = 16 724 W

$A_l = 0.0000664$ m^2

$AU_{cd} = 98\,000$ W K^{-1}

$\alpha = 0.01404$

$\dot{V}_{S_{FL}}$ 12 = 0.194611 m^3 s^{-1}

This procedure is a little time consuming since all the working points given by the manufacturer's catalog should be used as inputs of the identification routine. Moreover, the units used by this routine may differ from the units used in the catalog. But this identification has to be done only once and the units conversion can be handled by a worksheet software. The identified parameters can also be introduced in a database so that they can be used directly in an other project.
Since no information about part-load is given by the manufacturer's catalog, empirical relationships (Bourdouxhe et al., 1995) are used by the identification routine to estimate the part-load parameter, dp_{pump} . It is here equal to -15 011 Pa. This value does not have any physical meaning. The modeling of the part-load regime is in fact a bit too simplified.

5.2. The counterflow cooling tower

The manufacturer's catalog (B.A.C., 1988) allows the selection of a cooling tower on the basis of the supply air wet-bulb temperature, the water flow rate in the tower, the approach and the range. All these data, except the range, are defined directly by the sizing criteria. The required range can be determined thanks to the simulation of the selected chiller in design conditions. This simulation can be done once the chiller parameters are identified. It gives here a exhaust condenser water temperature equal to 34.5 °C and ,thus, a range equal to 5.5 °C in design conditions.

Once the cooling tower model is selected, the parameter identification can be done on the basis of one of its working point provided by the manufacturer's catalog. As it is shown in Figure 5, the parameter identification requires the supply air wet-bulb temperature, the water flow rate in the tower, the approach, the range, but also the air mass flow rate and the atmospheric pressure. Such as the first four data, the air volume flow rate is provided by the manufacturer's catalog. The air mass flow rate is deduced by assuming an air density of 1.15 kg m^{-3}. The atmospheric pressure is usually not taken into account by the manufacturer's catalog. It is therefore assumed to be equal to the standard pressure (101 325 Pa).
The cooling tower selected in the frame of this exercise is characterized by a fixed-speed fan

supplying an air volume flow rate equal to 10.57 m^3 s^{-1}. The identification procedure (Bourdouxhe et al., 1995; Bourdouxhe et al., 1994a) gives a global heat transfer coefficient equal to 23 018 W K^{-1} ($T_{wb_{su}}$ = 22 °C ; $\dot{M}_w$ = 20.66 kg s^{-1} ; Approach = 7.0 °C ; Range = 6.0 °C ; $\dot{M}_a$ = 12.16 kg s^{-1} ;
p_{atm} = 101 325 Pa).

5.3. The air-cooled reciprocating chiller

The information, for the full-load regime, given by the catalog of the air-cooled reciprocating chiller manufacturer (TRANE, Unknown date) is almost the same than the one found in the catalog of the water-cooled twin-screw chiller manufacturer. The difference is that the supply condenser air temperature is the only information needed to define the working conditions on the condenser side since the air flow rate of the condenser fans is imposed by the choice of the chiller model. The selection procedure is therefore similar to the one of the water-cooled twin-screw chiller.

Since the refrigerant used by the air-cooled chiller is also specified by the manufacturer's catalog (here, R22) and since the air flow rate can be replaced by a fictitious equivalent water flow rate (Equation (3)), the information still missing for the identification of the full-load parameters is the lower and upper limits of the condenser and evaporator heat transfer coefficients (see Figure 7). This information is used by the routine to optimize the identification procedure. However, the user has to grope for finding the value of these limits. The identification routine would be significantly improved if it could handle itself the determination of these limits.

Assuming that these limits are determined, the identification procedure (Bourdouxhe et al., 1995 ; Bourdouxhe et al., 1994b) can now be done and gives the following full-load parameters :

AU_{ev} = 61 783 W K^{-1}	AU_{cd} = 107 778 W K^{-1}
$\dot{W}_{lo}$ 13 = 9 880 W	α = 0.22984
C_f = 0.09991	$\dot{V}_{S_{FL}}$ 14 = 0.22719 m^3 s^{-1}

<u>Note</u> : The condenser air volume flow rate is equal to 52.8 m^3 s^{-1}
15
Such as for the water-cooled twin-screw chiller, this procedure is a little time consuming since all the working points given by the manufacturer's catalog should be used as inputs of the identification routine and since the units used by this routine may differ from the units used in the catalog. But, here again, this identification has to be done only once, the units conversion can be handled by a worksheet software and the identified parameters can be introduced in a database so that they can be used directly in an other project.
The only information about part-load given by the manufacturer's catalog is the steps of unloading. Although this information is needed by the simulation of the chiller in part-load regime, it is not sufficient for the identification of the part-load parameter, $\dot{W}_{pump}$. Empirical relationships (Bourdouxhe et al., 1995) are thus used by the identification routine to estimate the part-load parameter. $\dot{W}_{pump}$ is here equal to 15 000 W.

6. COMPARISON OF THE THERMAL PERFORMANCE OF BOTH COOLING PLANT DESIGN

Once the components of both cooling plants are selected and their parameters are identified, the simulation of the whole system can be approached.

The simulation user has first to define the auxiliary consumption's, the inputs required by the simulation, the control variables, the regulation rules and the assumptions considered in the simulation (e.g. uniform flow rate distribution).

In the first cooling plant design (characterized by the air-cooled chillers (see Figure 1)), the auxiliary consumption's are due to the condenser fans and the evaporator pumps. The condenser fan power consumption is given by the manufacturer's catalog (here, 21.6 kW). Contrary to the condenser fans, the evaporator pump power consumption can not be found in the chiller manufacturer's catalog since the chiller package does not contain the evaporator pump. It has then to be determined by the designer (let's consider here a pump power consumption of 8 kW).

In the second cooling plant design (characterized by the water-cooled chillers (see Figure 2)), the auxiliary consumption's are due to the condenser and evaporator pumps and the cooling tower fans. The cooling tower fan power consumption is given by the manufacturer's catalog (here, 7.5 kW). But the condenser and evaporator pump power consumption can not be found in the chiller manufacturer's catalog since the chiller package does not contain the condenser and evaporator pumps. It has thus to be determined by the designer (let's again consider here a pump power consumption of 8 kW).

The simulation requires two categories of data. The first category deals with the data that remain constant throughout the simulation or are imposed by the regulation (such as the component parameters, the pump and fan power consumption's and flow rate) while the second category deals with time-varying inputs. In the first design, these inputs are the building cooling loads and the outdoor air dry-bulb temperature. In the second design, these inputs are the building cooling loads and the outdoor air wet-bulb temperature. They should be known with a time basis of one hour. This time basis is commonly used to perform building system energy calculations.

In the first cooling plant design, the control variables are the number of operating chillers and the unloading of the compressors. These two variables are calculated so that the exhaust evaporator water temperature is equal to the value of the set point within a certain tolerance. Here, the set point temperature is equal to 6 °C and the tolerance is equal to 0.3 °C. Moreover, only one chiller is assumed to be used whenever the water temperature drop in the evaporator is lower than 2 °C. Since the manufacturer's catalog does not give any information about the regulation of the condenser fans, all of them are assumed to be used when the chiller is operating.

In the second cooling plant design, the additional control variables are the number of operating cooling towers and the bypass mass flow rate. These variables are controlled so that the supply condenser water temperature is always higher than a minimum value (here, 21 °C). Whenever possible, the bypass valve should be closed and an additional tower is switched on when the supply condenser water temperature is higher than a certain temperature (here, 28 °C).

Although such assumptions are used in practice, they do not necessarily lead to the optimal thermal performance of the cooling plant. This is a limitation to the use of “fixed“ control rules.

Once this preparation phase is completed, the implementation of the cooling plant simulation can be done.
The simulation software TRNSYS (Klein et al., 1994) is used in the frame of this exercise. Thanks to its modularity, it is quite simple to develop the simulation file of different design alternatives. However, this task remains time-consuming and could appear as forbidding for the practitioner. But one should realize that, in addition to their interest in the design stage, the simulation files can also be used in some other phases of the building life cycle (e.g. the commissioning). Wizards could be an efficient way to help the simulation user in creating the simulation files. This technique is notably widely used by the new text editor or worksheet software's.

The last step before the comparison of the thermal performance of the design alternatives is the implementation of the regulation rules. This step is certainly the most difficult and time-consuming step encountered by the simulation user in the HVAC system design stage. But no here again, the work done at this stage can also be used in other phases of the building life cycle. A possible solution to avoid the limitations inherent to "fixed" regulation rules as well as the implementation of a regulation routine, could be to compare automatically, at each simulation time step, the thermal performance of the cooling plant for each possible combination of the control variables (but, in exchange, it would increase significantly the calculation time).

Once the regulation rules are implemented, the simulation can be used to compare the design alternatives.
To compare efficiently both cooling plant design, the time-varying inputs should correspond to a test reference year. Nevertheless, for the sake of clarity, the paper will compare both designs on the basis of the design day (see Figure 10).

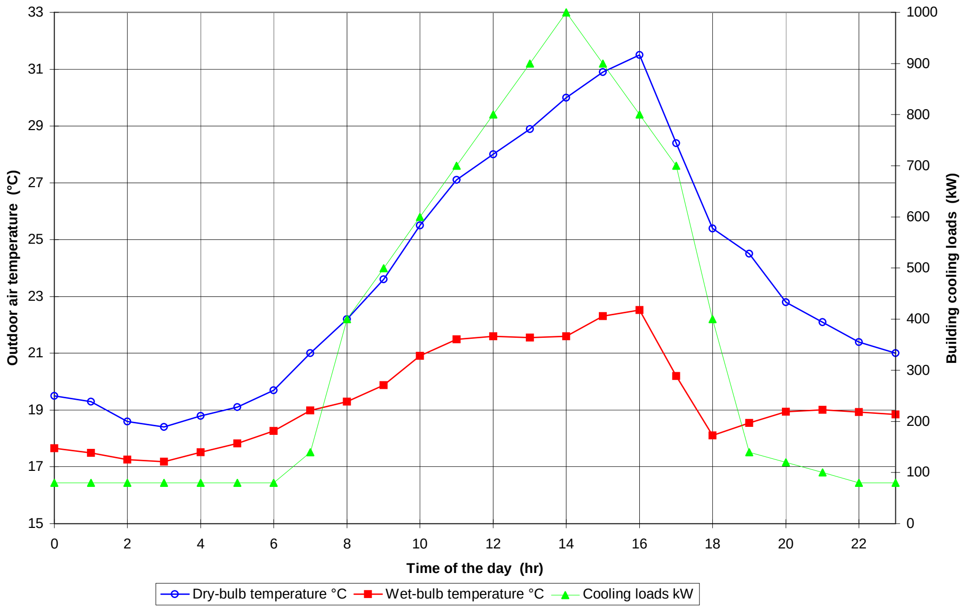

Figure 10 : Evolution of the building cooling loads and the outdoor dry-bulb and wet-bulb temperatures during the design day.

Figure 11 shows the evolution of the simulated exhaust evaporator water temperature and, for the second design, of the simulated supply condenser water temperature during the design day. As can be seen, both designs can respect the set point temperature (within the specified tolerance). This is logical since the components are sized to meet the maximum building cooling demand. The supply condenser water temperature also remains higher than the minimum temperature allowed by the regulation.

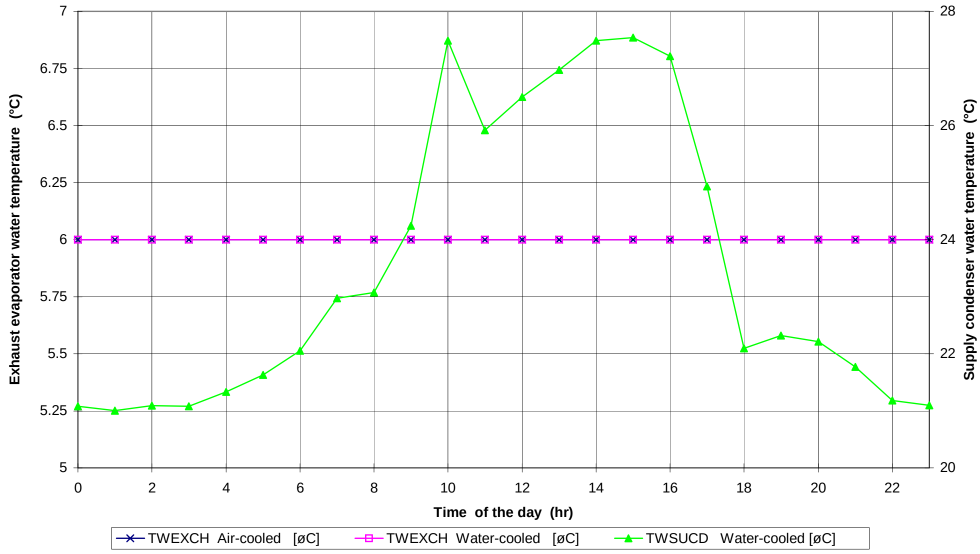

Figure 11 : Evolution of the exhaust evaporator water temperature and, for the second design, of the supply condenser water temperature during the design day

Figure 12 shows the evolution of the simulated number of operating chillers and, for the second design, of the simulated number of operating cooling towers and the simulated bypass mass flow rate. The operating status of the plant components defines the power consumption and, thus, the operating cost of the cooling plant. Figure 13 shows, for each design alternative, the evolution of the simulated chiller coefficient of performance as well as the simulated coefficient of performance of the whole systems.

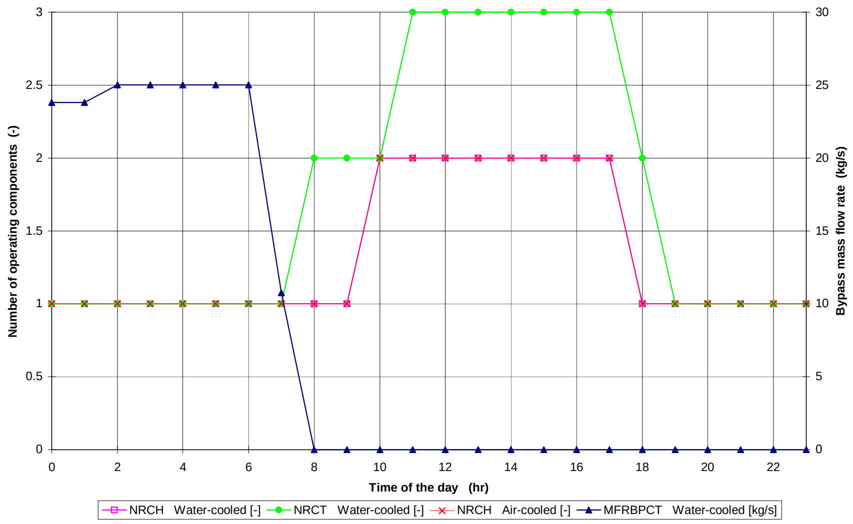

Figure 12 : Evolution of the number of operating chillers and, for the second design, of

the number of operating cooling towers and the bypass mass flow rate (design day)

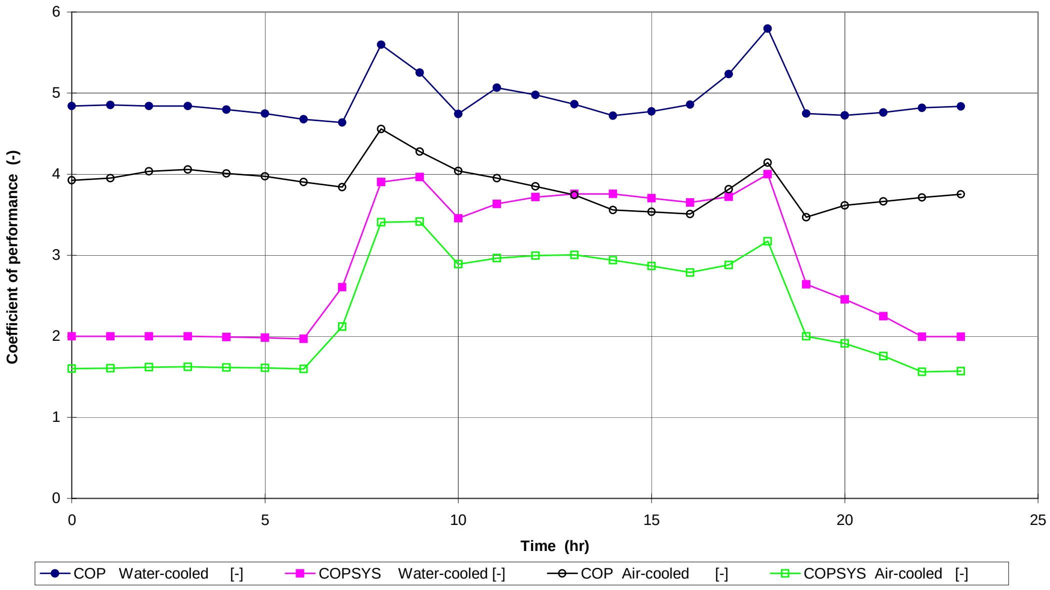

Figure 13 : Evolution of the chiller coefficient of performance and the coefficient of performance of the whole systems (design day)

The coefficient of performance of the whole systems accounts for the auxiliary consumption's. As can be seen in Figure 13, their influence is not negligible (at low building cooling loads, the coefficient of performance of the whole system can be 50 % lower than the chiller coefficient of performance) and they should be considered in the evaluation of the operating cost of the systems. Figure 13 also points out that the design based on water-cooled twin-screw chillers and cooling towers provides a better thermal performance than the design based on the air-cooled reciprocating chillers. The simulation over a test reference year could allow the designer to estimate the yearly operating cost benefits provided by the water-cooled chillers. This value could then help the designer in finding the design which represents the best compromise between the operating cost and the other design criteria (first cost, maintainability, etc.)

7. VARIATIONS OF THE SYSTEM CHARACTERISTICS

In addition to the comparison of design alternatives, the simulation can be used at the design stage to evaluate the effects of variations of the characteristics of the systems considered.
The system characteristics can be the sizing of the components, the value of the set points, the value of the fluid flow rates, etc. The value of the set points or of the fluid flow rates can be changed directly in the simulation file. If the designer wants to change the sizing of a component, he just has to identify the parameters of the new component on the basis of the manufacturer's catalog and use them in the simulation file. The identification procedure can even be avoided if the component parameters are already included in a database.

Figure 14 shows how an increase of 3 °C of the set point temperature at the evaporator exhaust can affect the coefficient of performance of the cooling plant characterized by the water-cooled chillers (the building cooling loads are assumed to remain unchanged).

As can be seen, the coefficient of performance of the cooling plant can be significantly improved by increasing the set point temperature at the evaporator exhaust (up to 10 % for a 3 °C increase). But, of course, this temperature increase also affects the thermal performance of the air handling units (at equal water mass flow rate, an increase of the supply cooling coil water temperature leads to a decrease of the cooling coil thermal performance). The set point temperature at the evaporator exhaust should therefore be chosen so that the air handling units be able to maintain the desired environmental conditions in the building. For example, the set point temperature could be related to the building cooling demand (it would increase as the building cooling loads decrease).

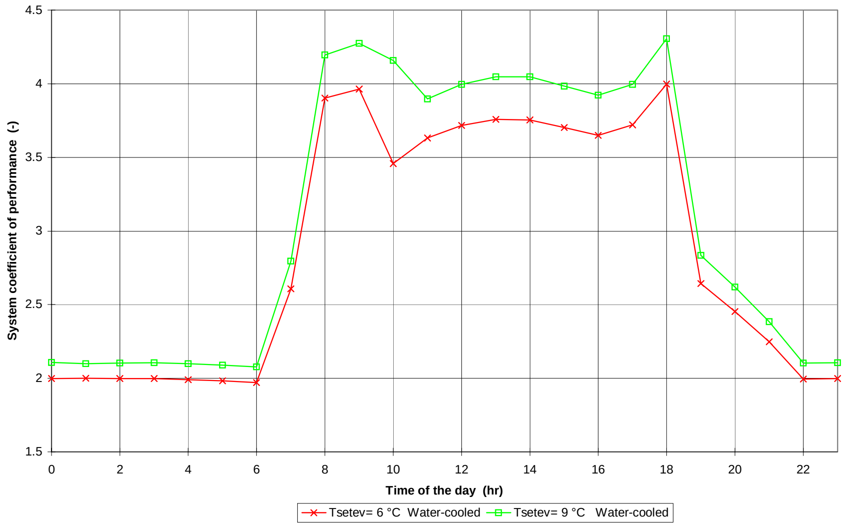

Figure 14 : Influence of the variations of the set point temperature at the evaporator exhaust (water-cooled chillers ; design day)

Figures 15 and 16 shows respectively how the cooling plant can respect the set point temperature at the evaporator exhaust (6°C) and how the coefficient of performance of the cooling plant is affected if the twin-screw chiller 1 is used instead of the twin-screw chiller 2 (see Table 1).

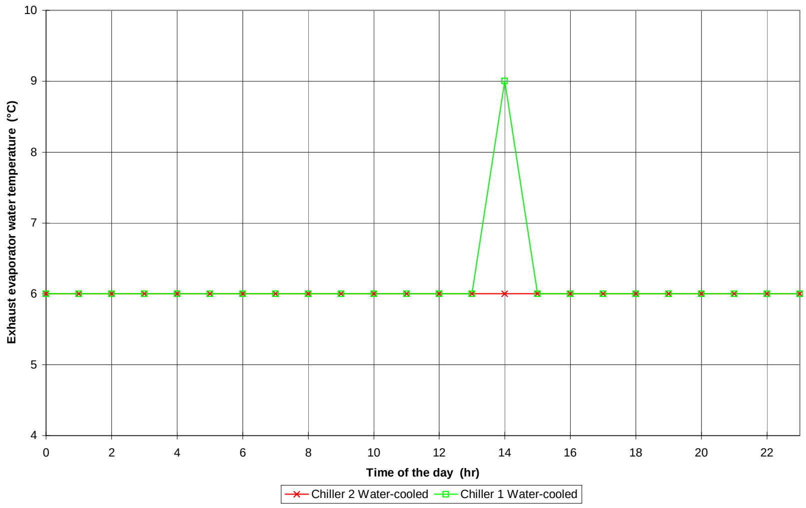

Figure 15 : Comparison of the exhaust evaporator water temperature provided by chillers of different capacities

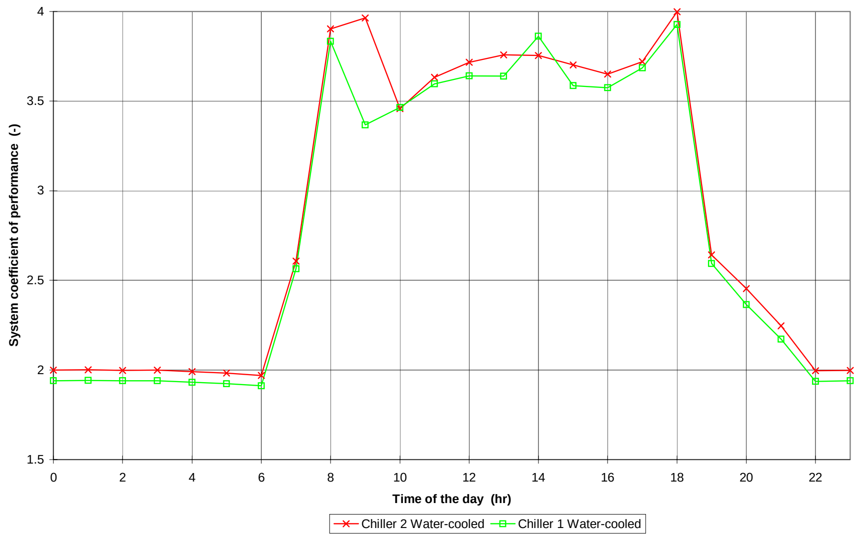

Figure 16 :Comparison of the system coefficient of performance provided by chillers of different capacities ($T_{set_{ev}}$ = 6 °C)

As can be seen in Figure 15, the cooling plant equipped with Chiller 1 (less powerful) is able to respect the set point temperature at the evaporator exhaust during the design day, except during the peak hour (14h00). During this hour, the cooling plant must allow a drift of the exhaust evaporator water temperature (till 9 °C) to be able to meet the building cooling demand.

Figure 16 shows that the coefficient of performance of the plant equipped with Chiller 1 is slightly lower than the coefficient of performance of the plant equipped with Chiller 2, except during two hours : at 09h00, the system coefficient of performance given by Chiller 1 is significantly lower since both chillers have to operate to respect the set point temperature at the evaporator exhaust, and at 14h00, the system coefficient of performance given by Chiller 1 is higher since the level of the water temperature in the evaporator is about 3 °C higher than the level of temperature characterizing the performance of Chiller 2.

Nevertheless, the results shown in Figure 16 may look surprising. Indeed, Chiller 1 is less powerful than Chiller 2. It has then to work closer to the full-load regime and, thus, closer to its optimal thermal performance. One could then expect a better coefficient of performance than with Chiller 2. Two reasons could explain this opposite tendency. The first one is that the evaporator and condenser of Chiller 2 are characterized by a higher global heat transfer coefficient. The second reason is the inaccuracy of the simulation in part-load regime : the parameter characterizing the part-load regime is just an estimation provided by empirical relationships. Therefore, the chiller thermal performance may not be correctly simulated when the chiller is operating at low regime.

Be that as it may, it seems that an undersizing of the chiller may not necessarily lead to a lower operating cost of the cooling plant (but, of course, the first cost of the undersized system is lower).

8. CONCLUSIONS

It emerges for this paper that the simulation can be used at the design stage of the primary equipment. The parameter identification of component models such as developed in the ASHRAE toolkits (Bourdouxhe et al., 1995) can deal with the information available at the design stage (i.e. with the data provided by the manufacturers' catalogs). The time-varying inputs required by the simulation can be given by a test reference year and by the building cooling loads calculation (this calculation should take the contribution of the cooling of the outside air into account).

The comparison of the thermal performance of both cooling plant alternatives shows the superiority of the cooling plant equipped with the water-cooled twin-screw chillers and the cooling towers. This comparison also shows that the auxiliary consumption's should be considered in the evaluation of the operating cost of the systems. However, although the simulation can allow the estimation of the yearly operating cost benefits provided by the design based on water-cooled chillers, it is up to the designer to find the design which represents the best compromise between the operating cost and the other design criteria (first cost, maintainability, etc.).

The interest of the simulation does not lie only in the comparison of design alternatives. It is also very convenient in the evaluation of the effects of variations of the characteristics of the systems considered (sizing of the components, value of the set points, value of the fluid flow rates, etc.).

Unfortunately, the use of the simulation software's also presents some limitations that can discourage the practitioner :

- there is a lack of information about the thermal performance of the components in part-load regime. This problem can be partially solved by the use of empirical relationships, but they may not be accurate enough when comparing different component sizing
- the consumption's of the evaporator and condenser pumps are not directly available
- some procedures are time consuming :
    - the first one is the identification of the chiller full-load parameters. But this identification has to be done only once and the identified parameters can be introduced in a database so that they can be used directly in an other project.
    - the second one is the development of the simulation files. But they can also be used in some other phases of the building life cycle (e.g. the commissioning). The simulation software interface could be improved to provide wizards that help the user in defining and modifying the system to be simulated.
    - the last, but not the least is the implementation of the regulation rules. This procedure has to be done to compare the design alternatives, but they do not necessarily lead to an optimized solution (especially when dealing with "fixed" regulation rules). Nevertheless, a possible solution to avoid the limitations inherent to "fixed" regulation rules as well as the implementation of a regulation routine, could be to compare automatically,

at each simulation time step, the thermal performance of the cooling plant for each possible combination of the control variables (but, in exchange, it would increase significantly the calculation time).

9. NOMENCLATURE

A_l	Equivalent leakage area of the twin-screw compressor	(m^2)
AU	Cooling tower global heat transfer coefficient	($W\ K^{-1}$)
AU_f	Fictitious global heat transfer coefficient	($W\ K^{-1}$)
AU_{ev}	Evaporator global heat transfer coefficient	($W\ K^{-1}$)
AU_{cd}	Condenser global heat transfer coefficient	($W\ K^{-1}$)
$c_{p_{moist\ air}}$	Moist air specific heat	($J\ kg^{-1}\ K^{-1}$)
c_w	Water specific heat	($J\ kg^{-1}\ K^{-1}$)
C_f	Clearance factor	(-)
COP	Chiller coefficient of performance	(-)
COPSYS	Coefficient of performance of the whole plant	(-)
dp_{pump}	Pressure jump encountered by the refrigerant diverted in the twin-screw compressor	(Pa)
$\dot{M}_a$ 16	Air mass flow rate	($kg\ s^{-1}$)
$\dot{M}_w$ 17	Water mass flow rate	($kg\ s^{-1}$)
$\dot{M}_{w-equivalent}$ 18	Fictitious water mass flow rate	($kg\ s^{-1}$)
MFRBPCT	Bypass water mass flow rate	($kg\ s^{-1}$)
NRCH	Number of operating chillers	(-)
NRCT	Number of operating cooling towers	(-)
p_1	Pressure at the supply of the compressor cylinder	(Pa)
p_2	Pressure at the exhaust of the compressor cylinder	(Pa)
TWSUCD	Supply condenser water temperature	(°C)
TWEXCH	Exhaust evaporator water temperature	(°C)
$T_{set_{ev}}$	Set point temperature at the evaporator exhaust	(°C)
$\dot{V}$ 19	Refrigerant volume flow rate	($m^3\ s^{-1}$)
$\dot{V}_{S_{FL}}$ 20	Compressor swept volume in full-load regime	($m^3\ s^{-1}$)
$\dot{W}$ 21	Power consumed by the compressor	(W)
$\dot{W}_{lo}$ 22	Constant part of the compressor electromechanical losses	(W)
$\dot{W}_{in}$ 23	Internal power of the compressor	(W)
$\dot{W}_{pump}$ 24	"Pumping" losses	(W)
α	Factor defining electromechanical losses proportional to the compressor internal power	(-)

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11. ACKNOWLEDGMENTS

This study is subsidized by The Gilbert Burnay Foundation managed by the ATIC - “Belgian Royal Technical Society of Heating, Ventilation and Related Technology Industry”.