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ABSTRACT

This paper presents a sizing study of the buffer tank in chilled water A/C systems. In order to find out the adequate sizing criteria for buffer tanks in these kind of installations, a review of different manufacturers' guidelines was carried out and it was concluded that there are three main operation parameters affected by the volume of the tank: the ON cycle time, the OFF cycle time, and the number of starts per hour of the chiller compressor. In order to better understand the influence of each parameter, a theoretical analysis was carried out where the impact of the building thermal load in different design criteria was also studied. After a thorough analysis of all the studied criteria, it was concluded that the minimum ON cycle time which is necessary to ensure that the oil returns to the compressor is the most critical criterion. Finally, a design guideline is proposed by the authors in order to determine the minimum volume of the buffer tank, mainly depending on the capacity of the chiller and the temperature deadband of the thermostat control.

Keywords

buffer tank; chiller operation; hydraulic design

NOMENCLATURE

Symbol

c_p	Specific heat at constant pressure = $4.18 kJ \cdot kg^{-1} \cdot K^{-1}$
n	Number of stages (compressors) running in a particular moment $(-)$
N	Total number of stages (compressors) of the chiller modules $(-)$
$\dot{q}$	Building thermal load (kW)
$\dot{Q}$	Total capacity of the chiller modules (.)
$\dot{Q}_1$	Unitary chiller capacity or compressor stage (kW)
t	Time (s)
T	Temperature (^oC)
$T_{o,ic}$	Temperature at the outlet of the internal circuit (^oC)
$T_{i,ic}$	Temperature at the inlet of the internal circuit (^oC)
T_{SB}	Building supply temperature (^oC)
sph	Compressor starts per hour $(-)$

V Volume of the system (m^3)

$\dot{V}$ Flow rate (m^3 / s)

[X_{min} to X_{max}] Value ranging from X_{min} to X_{max}

Subscripts

db Dead band

ON ON cycle

OFF OFF cycle

Greek symbols

α Partial load ratio of the system $(-)$

α_1 Partial load ratio of one stage $(-)$

ρ Water density = $1000\, kg{\cdot}m^{-3}$

Δ Difference

1.INTRODUCTION

HVAC systems in buildings account for a considerable percentage of the greenhouse gas emissions produced in developed countries [1]. Notwithstanding the importance of a good operation of chiller systems, a good performance of these systems starts with a proper design of all its components. Particularly, buffer tanks have always been employed in chiller installations in order to add adequate thermal inertia to the system. However, explanation about which is the influence of its volume or about how to size the tank is scarce and disperse.

Recent studies by Karlsson and Fahlén estimated the possible gains when using a buffer tank in order to better match capacity and load [2] and mentioned the use of a storage tank that makes it possible to keep the flows at their optimum [3]. Even though they also mentioned the influence of the thermal inertia of the system on the cyclic behavior of the chiller, none of them studied in depth the tank design. Other authors have analyzed the influence of the thermal inertia on the dynamic performance of these systems [4], but again they do not delve into the sizing of the buffer tank.

In previous studies the authors had already considered a buffer tank in order to provide thermal inertia and included it in a mathematical model of the system [5]. Nevertheless, it was not until the need arose within the framework of the GROUND-MED project [6] that efforts were focused on sizing the buffer tank.

In [7], the authors already employed the minimum ON time for an adequate compressor oil return (around 2 minutes) as the criterion to size the volume of the buffer tank.

Regardless of the publications commented above, little information has been found by the authors in the Technical Literature when it comes to the role and influence of the water tank in the system. Nevertheless, what the authors did find is scattered information concerning design guidelines and recommendations from HVAC equipment manufacturers and HVAC installers. The purpose of this work was not only to gather all that dispersed information and compare the different guidelines, but also to analyze the influence of the water tank volume on the energy performance of the system. Additionally, the influence of the different design criteria in the sizing of the tank was studied and the most restrictive approach was selected as the best design criterion.

2.SYSTEM DESCRIPTION

When it comes to the buffer tank and the piping circuit in chiller systems, there are different connection principles as shown in Figure 1: tank on the return line (Figure 1a), tank on the supply line (Figure 1b) and decoupling tank (Figure 1c). The thermal load of the building, $\dot{q}$, is heat added to the hydraulic loop, which is removed by the chiller modules which have a maximum cooling capacity $\dot{Q}$. All three configurations shown in Figure 1 are usual in small buildings and, although this paper focuses on the most common approach, which is locating the tank on the return line, the conclusions drawn are perfectly applicable for the rest of configurations.

It is common that this kind of systems do not run continuously, but stop during the night. That is the reason why it is not recommended to oversize too much the tank volume because, otherwise, more time and energy would be required to cool down a greater amount of water (if the tank is larger) when the installation starts in the morning due to the heat won during the night. On the other hand, the tank should be large enough as to provide the system with the necessary thermal inertia to couple the building thermal load with the chiller capacity and modulation. Therefore, an accurate determination of the necessary volume for the buffer tank is required in order to adjust its size as much as possible.

It should be clarified that, in a system as the one presented in Figure 1, the total volume of the system should be expressed as the volume of the buffer tank plus the volume of the piping circuit, where the piping circuit volume will depend on each installation (distance from the machinery room to the terminal units, pipes diameter, etc.): $V = V_{buffer\ tank} + V_{piping}$. The equations and calculations considered in the present work refer to the total volume of the system (V). However, in order to simplify the problem it is assumed that all the volume of the system is concentrated in the buffer tank and that the piping volume is negligible. As a consequence, in order to obtain the volume of the actual buffer tank from the formulae developed in this paper, the volume of the piping circuit must be substracted from the calculated system volume (V).

3.SIZING THE BUFFER TANK

As it will be presented below, the authors found from the Literature review carried out that the main parameters to take into consideration when undertaking the design of the buffer tank are the minimum operating time of the compressor ($\Delta t_{min\ ON}$), the minimum OFF cycle time ($\Delta t_{min\ OFF}$) and the maximum number of starts per hour of the compressor (sph_{max}). Therefore, the influence of these three parameters in the sizing of the buffer tank is analysed in the present work following two different approaches: a theoretical one based on a simplified energy balance analysis and a second one regarding practical considerations.

3.1 THEORETICAL APPROACH

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In a system as the one presented in Figure 1a, an energy balance can be separately performed for the ON cycle time (time during which the compressor (stage) switching between ON/OFF is working) and for the OFF cycle time (time during which the compressor is switched off). Considering n as the number of compressor stages running in a particular moment to meet the demand of the building, during the OFF period of compressor n, there will be only $n-1$ compressors running. In that case, the active cooling capacity will be equal to $(n-1)\cdot\frac{\dot{Q}}{N}$, assuming each stage with the same unitary capacity $\left(\dot{Q}_1 = \frac{\dot{Q}}{N}\right)$, where N is the total number of stages of the compressor rack or chiller modules. Therefore, the energy conservation equation along the OFF period of compressor n assuming that all the water is concentrated in the bufer tank can be expressed in the following way:

$$\rho V \cdot c_p \cdot \frac{dT}{dt} = \dot{q} - (n-1)\cdot\frac{\dot{Q}}{N} \quad (1)$$

Where: $\dot{q}$ is the building thermal load,

$\rho\cdot V$ is the mass (m) of water in the system,

c_p is the specific heat, and

$\frac{dT}{dt}$ is the tank (system) water temperature variation with time.

The instantaneous thermal load of the building, $\dot{q}$, could be expressed as a fraction of the total chiller plant capacity, $\dot{q} = \alpha\cdot\dot{Q}$, where α stands for the partial load ratio of the system (In a good usual design, the value of α will therefore be slightly below 1). The partial load ratio of the system, $\alpha = \frac{\dot{q}}{\dot{Q}} = \frac{\dot{q}}{N \cdot \dot{Q}_1}$, should be distinguished from that of each individual stage,

$\alpha_1 = \frac{\dot{q} - (n-1)\dot{Q}_1}{\dot{Q}_1}$, being both of them related through the following expression: $\alpha = \frac{\alpha_1 + (n-1)}{N}$

, which can be also expressed as $\alpha = (1-\alpha_1) \cdot \frac{n-1}{N} + \alpha_1 \cdot \frac{n}{N}$. Taking into account the above considerations, equation (1) can be easily solved for the OFF cycle time (Δt_{OFF}) leading to equation (2), where ΔT_{db} is the temperature increase experienced by the water in the system during the OFF time, which is equal to the temperature deadband set in the chiller control.

$$\Delta t_{OFF} = \frac{\rho \cdot V \cdot c_p \cdot \Delta T_{db}}{\dot{q} - (n-1) \cdot \frac{\dot{Q}}{N}} = \frac{\rho \cdot V \cdot c_p \cdot \Delta T_{db}}{\alpha_1 \cdot \dot{Q}_1} \quad (2)$$

Using equation (2), the minimum volume required for the system to provide a minimum OFF time can be estimated by means of equation (3).

$$V_{min} = \frac{\alpha_1 \cdot \dot{Q}_1}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \Delta t_{min\,OFF} \quad (3)$$

The same analysis can be performed for the ON cycle time, where the chiller capacity increases up to $n \cdot \frac{\dot{Q}}{N} = n \cdot \dot{Q}_1$ and the temperature will decrease following Equation (4):

$$\rho \cdot V \cdot c_p \cdot \frac{dT}{dt} = \dot{q} - n \cdot \dot{Q}_1 \quad (4)$$

Proceeding as previously presented for the OFF time period, and taking into account that the temperature variation will be negative ($-\Delta T_{db}$), the ON cycle time is provided by equation (5).

$$\Delta t_{ON} = \frac{\rho \cdot V \cdot c_p \cdot \Delta T_{db}}{(1-\alpha_1) \cdot \dot{Q}_1} \quad (5)$$

Therefore, the minimum volume of the system required to provide a minimum ON time is presented in equation (6).

$$V_{min} = \frac{(1-\alpha_1) \cdot \dot{Q}_1}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \Delta t_{min\,ON} \quad (6)$$

Finally, the minimum volume required for the system in order to avoid a maximum 'number of starts per hour' (sph) of one compressor stage can be obtained by using the total duration of a complete cycle ($\Delta t = \Delta t_{ON} + \Delta t_{OFF}$). By definition, the number of starts per hour is the number of complete cycles in an hour ($3600\,sec$) as presented in equation (7).

$$sph = \frac{3600}{\Delta t_{OFF} + \Delta t_{ON}} \quad (7)$$

Combining equations (2), (5) and (7), equation (8) can be obtained, which provides the number of starts per hour as a function of the partial load ratio of the system.

$$sph = \frac{3600}{\frac{\rho \cdot V \cdot c_p \cdot \Delta T_{db}}{\dot{Q}_1} \cdot \left(\frac{1}{\alpha_1} + \frac{1}{1-\alpha_1} \right)} = \frac{3600 \cdot \dot{Q}_1 \cdot \alpha_1 \cdot (1-\alpha_1)}{\rho \cdot V \cdot c_p \cdot \Delta T_{db}} \quad (8)$$

Then the minimum volume of the system required to avoid a maximum number of starts per hour of one compressor stage can be estimated by means of equation (9).

$$\mathrm{V}_{min} = \frac{3600}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \frac{\alpha_1 \cdot (1-\alpha_1) \cdot \dot{Q}_1}{sph_{max}} \quad (9)$$

From the energy balance analysis presented above, three different expressions to calculate the minimum volume of the system were obtained according to three different criteria: $\Delta t_{min\,ON}$,

$\Delta t_{min\,OFF}$ and sph_{max}, according to equations (3), (6) and (9) respectively. As it can be observed, the minimum system volume is in all cases proportional to the capacity of one stage ($\dot{Q}_1$) and to the inverse of the temperature deadband. For compressor racks or chiller modules with different capacity stages $\dot{Q}_1$ represents the largest capacity step of the system modulation.

Additionally, equations (3), (6) and (9) depend on the partial load of the stage, α_1, which can vary from 0 to 1 depending on the building thermal load.

As an example, Figure 2 depicts, for all three criteria, the minimum volume that the system would require as a function of the partial load ratio, for the case of a chiller of 20 kW with a single compressor (number of stages n = 1) $\dot{Q} = \dot{Q}_1 = 20kW$; and the following typical values of the other parameters: $\Delta T_{db} = 3K$; $\Delta t_{min\,OFF} = 3min$; $\Delta t_{min\,ON} = 2min$; $sph_{max} = 10$.

As it can be observed in Figure 2, when considering the minimum ON cycle time for the design (equation (6)), the minimum volume of the system is directly proportional to $(1-\alpha_1)$, what means that the lower the partial load ratio, the higher the minimum volume. Conversely, with the criterion of the minimum OFF cycle time, the minimum system volume would be directly proportional to the partial load ratio (equation (3)). Therefore, the higher the load ratio, the higher the minimum volume required. Finally, when it comes to the number of starts per hour of the compressor, the volume is quadratically dependant on the partial load ratio (equation (9)), hence presenting a maximum which comes for $\alpha_1 = 0.5$ independently of the values of the other parameters. So, the closer the load ratio to the value of 0.5, the higher the minimum volume required for the system. As it can be observed in this example, the most restrictive criterion would correspond to the minimum OFF cycle time, being the minimum total volume required equal to $0.287\,m^3\left(287\,L\right)$.

Given that the partial load ratio per stage α_1, varies by definition from 0 to 1, the following expressions can be obtained for the minumum value of the volume corresponding to the three considered criteria.

$$V_{min} = \frac{\alpha_1 \cdot \dot{Q}_1}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \Delta t_{min\,OFF} = \frac{\dot{Q}_1}{\rho \cdot c_p} \cdot \frac{\Delta t_{min\,OFF}}{\Delta T_{db}} \qquad (10)$$

$$V_{min} = \frac{\left(1-\alpha_1\right) \cdot \dot{Q}_1}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \Delta t_{min\,ON} = \frac{\dot{Q}_1}{\rho \cdot c_p} \cdot \frac{\Delta t_{min\,ON}}{\Delta T_{db}} \quad (11)$$

$$V_{min} = \frac{3600}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \frac{\alpha_1 \cdot \left(1-\alpha_1\right) \cdot \dot{Q}_1}{sph_{max}} = \frac{\dot{Q}_1}{\rho \cdot c_p} \cdot \frac{\frac{900}{sph_{max}}}{\Delta T_{db}} \quad (12)$$

Equations (10), (11) and (12) provide, from a theoretical point of view and considering three different criteria, the minimum volume required for the system as a function of the unitary chiller capacity (or largest capacity step of the system modulation) $\dot{Q}_1$.

3.2 MANUFACTURERS' GUIDELINES REVIEW

HVAC equipment manufacturers often publish guidelines on how to operate systems and how to size different components in order to achieve a better performance of their units. Likewise, installers usually give recommendations based on their broad practical experience. Some of these guidelines and recommendations when addressing the sizing of the buffer tank have been gathered by the authors and are presented and discussed in the following subsections.

3.2.1 Popular design criterion

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According to some HVAC engineering companies and installers consulted, the most general indication when facing buffer tank sizing is that "*the tank must supply the chilled water during the necessary off-time of the compressor*." This is analytically represented by equation (13), where ΔT_{db} is the deadband of the thermostat control, which is typically around 2 - 3K.

$$\rho \cdot V \cdot c_p \cdot \Delta T_{db} = \dot{Q}_1 \cdot \Delta t_{\min OFF} \quad (13)$$

Notice that it is the unitary capacity of the chiller (or compressor stage) $\dot{Q}_1$ what is considered, and not the total capacity of the chillers plant ($\dot{Q}$).

Therefore, there is a minimum volume of the tank which allows the system to meet the demand when the corresponding stage of the chiller is switched off (OFF period). This minimum volume is provided by equation (14). In order to make a later comparison, this approach will be referred to as "POPULAR."

$$V_{min} = \frac{\Delta t_{\min OFF}}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \dot{Q}_1 \quad (14)$$

3.2.2 ASHRAE recommendations

In Chapter 10 '*The Design, Construction and Operation of Sustainable Buildings'* of the ASHRAE green guide [8], The following recommendation appears: "*Manufacturers of water chillers state that system water volume should be a minimum of 3 to 10 gal per installed ton of cooling (0.054 to 0.179 L per kWR [kW of Refrigeration]). In a system less than this and under light cooling load conditions, thermal inertia coupled with the reaction time of chiller controls may cause the units to short cycle or shut down on low-temperature safety control.*." In other

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words, $V_{min} = \left[3 to 10\right] gal / cooling\,ton$. After converting to International System units, equation (15) is obtained.

$$V_{min} = \left[3.24 to 10.8\right]\cdot\dot{Q}_1 \quad (15)$$

3.2.3 CARRIER / INTERCLISA recommendations

An interesting explanation for the matter was found in the following reference [9]. In this reference, it is argued that the minimum liquid volume for chiller plant installations should be that which "*just gives time to the temperature sensors to track temperature variations*", recommending that the temperature variation should be lower than $1.5\,^{o}C$ per minute, i.e. $\left|dT/dt\right| \le 1.5\,^{o}C/min = 1.5/60\,K/s = 0.025\,K/s$. Taking into account that the maximum variation of the temperature will happen when a full unitary chiller capacity step (or largest capacity step of the system modulation) $\dot{Q}_1$ is applied to the water circuit, the following formula is obtained:

$$\rho\cdot V\cdot c_p\cdot\left.\frac{dT}{dt}\right|_{max} = \dot{Q}_1 \qquad (16)$$

when n stages are switched on.

Therefore, the minimum volume of the system would be given by equation (17).

$$V_{min} = \frac{\dot{Q}_1}{\rho\cdot c_p\cdot\left.\frac{dT}{dt}\right|_{max}} \qquad (17)$$

For a later comparison this approach will be called "CARRIER(1)."

An alternative approach considered in the same reference (from now on referred to as "CARRIER(2)") takes into account the "*minimum off-cycle time for one compressor partialization step.*" This is analytically expressed by equation (18).

$$\rho \cdot V \cdot c_p \cdot \frac{\Delta T_{db}}{\Delta t_{min\,OFF}} = \dot{q} - \frac{n}{N} \cdot \dot{Q} = \dot{q} - n \cdot \dot{Q}_1 \quad (18)$$

This expression is very interesting and clearly recognises that temperature variations are induced by load steps of the compressors rack. The maximum variation would take place when the building load tends to be slightly higher than the previous capacity step $(n-1) \cdot \dot{Q}_1$. Accordingly, equation (18) transforms into equation (19), which is exactly the same as the POPULAR criterion.

$$V_{min} = \frac{\dot{Q}_1}{\rho \cdot c_p \cdot \frac{\Delta T_{db}}{\Delta t_{min\,OFF}}} \quad (19)$$

In the same reference, the author recommends a value of 5 minutes for the minimum $\Delta t_{min\,OFF}$.

3.2.4 Trane recommendations

The authors have found recommendations on this regard by Trane in the technical manuals of some of their chillers. They distinguish somewhat between high capacity (e.g. Model CGAF [15-60 ton] (53-211 kW)) and small capacity (e.g. Model CGA [10-15 ton] (35-53 kW)) units [10]. They state the following: "*The volume of water in the loop is critical to the stability of system operation. The minimum required water volume is dependant on the chiller controller and system GPM. Water volumes less than the minimum required for the system can cause nuisance problems including low pressure trips and freezestat trips. The cause of these trips is "Short*

Water Loops". The minimum required water volume (as a function of loop time and GPM) is as follows:

CGAF: Minimum Loop Volume = GPM x 3 Minute Loop Time

CGA: Minimum Loop Volume = GPM x 5 Minute Loop Time"

For larger units (up to 120 ton (422 kW)) they recommend that "*a two-minute water loop circulation time is sufficient to prevent short water loop issues*" [11].

In summary, Trane recommends different loop times depending on the size of the unit:

- 10 to 15 ton (35-53 kW): Minimum Loop Volume = GPM x 5 min Loop Time (TRANE(1))
- 15 to 60 ton (53-211 kW): Minimum Loop Volume = GPM x 3 min Loop Time (TRANE(2))
- Up to 120 ton (422 kW): Minimum Loop Volume = GPM x 2 min Loop Time (TRANE(3))

Where GPM is the flow rate of water in gallons per minute.

Taking into consideration the above recommendations, it is clear that the minimum volume would depend on the flow rate and the total loop time. In other words, $V_{min} = \dot{V} \cdot \Delta t$.

In contrast with previous recommendations, it is interesting to find that this time the recommendation of minimum volume is referred to the water flow rate. However, if one takes into account that the total water flow rate, $\dot{V}$, is related to the total chiller capacity, $\dot{Q}$, through the following relation, $\dot{V} = \dot{Q} / \left(\rho \cdot c_p \cdot \Delta T_{chiller} \right)$, where $\Delta T_{chiller}$ is the water temperature difference across the chiller (chillers in parallel), the formula for the minimum volume can also be casted in the form of previous expressions, as shown in equation (20).

$$V_{min} = \frac{\dot{Q}}{\rho \cdot c_p \cdot \Delta T_{chiller}} \cdot \Delta t \quad (20)$$

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where the time period, Δt, would depend on the size of the system, ranging from 2 to 5 minutes. The larger the size, the shorter the period. It is important to notice that, in the case of this reference, the capacity considered in the expression to calculate the minimum system volume corresponds to the total chiller modules capacity ($\dot{Q}$) and not to the unitary capacity of the chiller or compressor stage ($\dot{Q}_1$).

3.2.5 Danfoss Commercial Compressors (DCC) recommendations

DCC state the following guidelines for their scroll compressors [12]:

−*The system must be designed in a way that guarantees a minimum compressor running time of 2 minutes so as to provide for sufficient motor cooling after start-up along with proper oil return*

−*There must be no more than 12 starts per hour; a number higher than 12 reduces the service life of the motor-compressor unit.*

−*A minimum three-minute (180 s) time out is recommended.*

Summarizing, with the nomenclature employed in this paper, the recommendations are as follows:

$-\Delta t_{min\,ON} \geq 2\,min\left(or\,oil\,return\,time\right)$

$-\Delta t_{min\,OFF} \geq 3\,min$

$-Starts\,per\,hour\left(sph\right) \leq 12$

3.2.6 EMERSON recommendations

Emerson Climate Technologies states the following guidelines for their Copeland scroll compressors [13]:

−*There must be a maximum of 10 starts per hour.*

−*There is no minimum off time because Scroll compressors start unloaded even if the system has unbalanced pressures.*

−*The most critical consideration is the minimum run time required to return oil to the compressor after start-up.*

Summarizing, with the nomenclature employed in this paper, the recommendations are as follows:

$-\Delta t_{min\,ON} \geq Oil\,return\,time$

$-\Delta t_{min\,OFF} - No\,limit\ for\,scroll\,compressors$

$-Starts\ per\,hour\left(sph\right) \leq 10$

3.2.7 Summary of guidelines and recommendations

The following conclusions can be drawn from the analysis of Table 1:

−On the one hand, HVAC system manufacturers provide design guidelines to determine the minimum volume of the system. On the other hand, compressor manufacturers just give recommended values for minimum ON and OFF cycle times as well as maximum number of starts per hour of the compressor, as the refrigeration circuit is unknown for them. That is the reason why they only provide recommendations for the compressor operation.

−All the expressions for the minimum volume of the system (buffer tank + piping circuit) can be expressed as a function of the chiller unitary capacity or compressor stage ($\dot{Q}_1$), which obviously is the main parameter for its sizing, with the exception of Trane guidelines. In the case of Trane recommendations, it is the total chiller modules capacity ($\dot{Q}$) that is considered instead of the unitary capacity of the chiller. Since this paper focuses on small-medium installations, only the recommendation for small capacity units (TRANE(1)) will be considered.

−As discussed in section 3.1, the time periods are a function of the system volume and capacity stage, but also of the setting of the temperature control deadband. The criteria POPULAR and CARRIER(2) also include this parameter. The typical value of the temperature deadband employed by manufacturers is 2K. However, most of the time the temperature probe of the thermostat is externally clamped to the pipe wall, what in practice means that the actual temperature variation of the water flow at the point of control is larger. The authors have accurately measured this difference in some installations and a typical setting of 2K as the control deadband transforms into a 3K to 4K temperature variation in the actual water flow.

Taking into account that energy minimization requires to minimize the temperature difference between the chilled water and the ambient air, this would require a more precise control of the supply temperature and therefore small values of the control deadband. Obviously, this would require larger buffer tanks. A typical value of 3K for the actual variation of the water temperature in the control point has been assumed as a good reference value for the rest of the following comparisons. In any case, this parameter will be retained as a second input to the formula since it is a control parameter and can be set at the installation.

−In regard to the other parameters, some of the manufacturers provide recommended values for them. However, in the following cases they need to be assumed:

- The temperature difference to be considered in "TRANE" formulae is the one between the chiller inlet and outlet. The typical value employed by designers for the nominal flow rate is $\Delta T_{chiller} = 5K$.
- "POPULAR" requires an estimation of the minimum OFF time. The value employed in this paper will be the one recommended by DCC, that is $\Delta t_{minOFF} = 3min$.

4.RESULTS AND DISCUSSION

Table 2 summarizes the results for the expressions obtained from both the theoretical approach in section 3.1 and the different criteria extracted from the Literature review in section 3.2, after considering the values corresponding to the density and specific heat of the water as well as the recommended values for the operation parameters ($\Delta t_{\min OFF}$, $\Delta t_{\min ON}$, sph_{max} *and* ΔT_{db}). For the theoretical approach criteria, the following values have been used: $\Delta t_{\min OFF} = 3min$; $\Delta t_{\min ON} = 2min$; $sph_{max} = 10$; $\Delta T_{db} = 3K$. For the manufacturers' guidelines criteria, the values from Table 1 have been considered. The last column in Table 2 shows the resulting minimum volume, in liters (L), for the used example of a 20 kW chiller with a single compressor ($\dot{Q} = \dot{Q}_1 = 20kW$).

After comparing the results obtained in Table 2, it can be observed that the most restrictive criterion regarding the maximum system volume is "CARRIER(2)." This is because an OFF time of 5 minutes has been considered, which might be excessive according to the authors since it almost doubles the resulting minimum volume recommended by the rest of the reviewed guidelines. Apart from this, the rest of criteria range from 3.24 to 14.35 times the value of the chiller capacity. There are several sources with coincident results and, on the whole, all of them present relatively similar values. Therefore, a possible solution for the tank sizing might be considering, as a rule of thumb, equation (21), which would be conservative enough and cover most of the reviewed guidelines for small and medium installations.

$$V_{min}\left(liters\right)=\left[7\,to\,14\right]\cdot\dot{Q}_1\left(kW\right) \qquad (21)$$

However, it is worth following discussing the obtained formula and trying to develop a more accurate final design guideline. As it can be seen in Table 2, the three analytical expressions developed in section 3.1 take into account the most important requirements for the sizing of the buffer tank volume ($\Delta t_{\min OFF}$, $\Delta t_{\min ON}$ and sph_{max}). Considering these expressions, the most restrictive approach appears to be the minimum OFF time criterion. Nevertheless, this does not seem to be so important for scroll compressors (see section 3.2.6). In contrast, what is clearly important to accomplish, and it is stated by all compressors manufacturers, is a minimum ON cycle time in order to guarantee that the oil returns to the compressor.

In any case, studying carefully these analytical expressions (equations (10), (11) and (12)), it is clearly observed that all of them present the same form. Moreover, the minimum volume will be provided in the end by the worst-case compressor time ($\Delta t_{compressor}$) considered in the numerator, whether this is the minimum OFF time, the minimum ON time or the time given by the maximum number of starts per hour of the compressor. Therefore, considering the values of the water density ($1000\,kg{\cdot}m^{-3}$) and the specific heat ($4.18\,kJ{\cdot}kg^{-1}{\cdot}K^{-1}$) and expressing the result of the volume in 'L', the time in 'min' and the capacity in 'kW', equation (22) provides the final design guideline recommended by the authors in order to calculate the minimum system volume required.

$$V_{min}\left(liters\right)=14.35\cdot\frac{\Delta t_{compressor}\left(min\right)}{\Delta T_{db}\left(^{\circ}C\right)}\cdot\dot{Q}_1\left(kW\right) \quad (22)$$

Where the compressor time, in minutes, is given by equation (23).

$$\Delta t_{compressor}\left(min\right)=\max\left(15/sph,\Delta t_{\min OFF},\Delta t_{\min ON}\right) \quad (23)$$

The authors recommend deadband values between 3 and 5 K ($\Delta T_{db}=3 to 5 K$), and compressor time values between 3 and 5 minutes ($\Delta t_{compressor}=3 to 5 min$). Even though 3 minutes is time enough to guarantee a proper compressor oil return for most compressors, a longer time (5 minutes) would produce longer cycles thus avoiding possible short water loops issues as well as reducing the losses due to low partial load.

It should be noticed that in installations with chiller modules provided with variable speed compressors, the unitary capacity ($\dot{Q}_1$) to be considered is the one corresponding to the ON/OFF at the minimum speed, so that it is clear that installations with regulation by variable speed will required a very small buffer tank if any.

5.CONCLUSIONS

The sizing study of the buffer tank in chilled water A/C systems was carried out, and it was concluded that there are three main operation parameters affecting the required volume for the system (piping circuit plus buffer tank): the ON cycle time, the OFF cycle time, and the number of starts per hour of the cycling chiller/compressor, which consistently are the three inputs for its sizing. A simple theoretical study allowed characterizing the sizing of the volume with simple formulae. In addition, a detailed review of manufacturers' guidelines and installers' recommendations was carried out. After a thorough analysis of all the studied criteria, it was concluded that both chiller and compressor manufacturers gave design guidelines that are mainly based on the compressor operation.

For instance, in the case of scroll compressors, there is no need for a minimum OFF cycle time since they start unloaded even if the system has unbalanced pressures, as stated by compressor manufacturers. Regarding the number of starts per hour, it would obviously affect the life span of the compressors, but it is not the most critical parameter as concluded in this work. Finally, the minimum ON cycle time turns out to be probably the most critical criterion in order to ensure that the oil returns to the compressor.

After a comprehensive analysis, the authors reached to a final rational design formula (equation (22)) in order to obtain the minimum required volume of the system as a function of the temperature control deadband, the unitary chiller capacity or compressor stage, and the time provided by the worst case condition among the three considered time requirements (equation (23)): the minimum OFF time ($\Delta t_{\min OFF}$), the minimum ON time ($\Delta t_{\min ON}$) and the maximum number of start per hour of the compressor (sph_{max}).

A rule of thumb was also proposed which states that the minimum system volume, in liters, should range from 7 to 14 times the unitary chiller capacity or compressor stage expressed in kilowatts (equation (21)).

It should be reminded that, in order to determine the buffer tank volume, the volume of the piping circuit must be subtracted from the one calculated by applying these design guidelines since the above mentioned formulas referred to the total water volume in the distribution loop.

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Table 1 shows a summary of the compiled guidelines and recommendations.

	Expression	$\Delta t_{min\ ON}$	$\Delta t_{min\ OFF}$	sph_{max}	Other
POPULAR	$V_{min} = \frac{\Delta t_{minOFF}}{\rho \cdot c_p \cdot \Delta T} \cdot \dot{Q}_1$	n/a	n/a	n/a	ΔT_{db} Δt_{minoff}
ASHRAE	$V_{min} = [3.24 to 10.8] \cdot \dot{Q}_1$	n/a	n/a	n/a	n/a
CARRIER(1)	$V_{min} = \frac{1}{\rho \cdot c_p \cdot \left.\frac{dT}{dt}\right\|_{max}} \cdot \dot{Q}_1$	n/a	n/a	n/a	$\left.\frac{dT}{dt}\right\|_{max} = 1.5 \frac{K}{min}$
CARRIER(2)	$V_{min} = \frac{1}{\rho \cdot c_p \cdot \frac{\Delta T}{\Delta t_{minOFF}}} \cdot \dot{Q}_1$	n/a	5 min	n/a	$\Delta T_{db} = 3K$
TRANE(1) (small)	$V_{min} = \frac{\Delta t}{\rho \cdot c_p \cdot \Delta T} \cdot \dot{Q}$	n/a	n/a	n/a	$\Delta T_{chiller}$ $\Delta t = 5 min$
TRANE(2) (medium)	$V_{min} = \frac{\Delta t}{\rho \cdot c_p \cdot \Delta T} \cdot \dot{Q}$	n/a	n/a	n/a	$\Delta T_{chiller}$ $\Delta t = 3 min$
TRANE(3) (large)	$V_{min} = \frac{\Delta t}{\rho \cdot c_p \cdot \Delta T} \cdot \dot{Q}$	n/a	n/a	n/a	$\Delta T_{chiller}$ $\Delta t = 2 min$
DCC	n/a	2 min	3 min	12	n/a
EMERSON	n/a	Oil return time	No limit for scroll comp.	10	n/a

Table 2. Minimum system volume in terms of the chiller capacity (example use of a 20 kW chiller with a single compressor)

Theoretical Approach	**Criterion**	**Expression**	$\boldsymbol{V_{min}[L]=f(\dot{Q}_1[kW])}$	**For** $\boldsymbol{\dot{Q}_1 = 20\ kW}$
	$\Delta t_{min\,OFF}$	$V_{min}=\frac{\Delta t_{min\,OFF}}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \dot{Q}_1$	$V_{min}=14.35 \cdot \dot{Q}_1$	$287.1L$
	$\Delta t_{min\,ON}$	$V_{min}=\frac{\Delta t_{min\,ON}}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \dot{Q}_1$	$V_{min}=9.57 \cdot \dot{Q}_1$	$191.4L$
	sph_{max}	$V_{min}=\frac{\frac{900}{sph_{max}}}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \dot{Q}_1$	$V_{min}=7.18 \cdot \dot{Q}_1$	$143.5L$
Manu-facturers' guidelines	POPULAR	$V_{min}=\frac{\Delta t_{min\,OFF}}{\rho \cdot c_p \cdot \Delta T_{db}} \cdot \dot{Q}_1$	$V_{min}=14.35 \cdot \dot{Q}_1$	$287.1L$
	ASHRAE	$V_{min}=[3.24 to 10.8] \cdot \dot{Q}_1$	$V_{min}=[3.24 to 10.8] \cdot \dot{Q}_1$	$[64.8 to 216]L$
	CARRIER(1)	$V_{min}=\frac{1}{\rho \cdot c_p \cdot \left.\frac{dT}{dt}\right\|_{max}} \cdot \dot{Q}_1$	$V_{min}=9.57 \cdot \dot{Q}_1$	$191.4L$
	CARRIER(2)	$V_{min}=\frac{1}{\rho \cdot c_p \cdot \frac{\Delta T_{db}}{\Delta t_{minOFF}}} \cdot \dot{Q}_1$	$V_{min}=23.9 \cdot \dot{Q}_1$	$478.5L$
	TRANE(1)	$V_{min}=\frac{\Delta t}{\rho \cdot c_p \cdot \Delta T_{chiller}} \cdot \dot{Q}$	$V_{min}=14.35 \cdot \dot{Q}$	$287.1L$

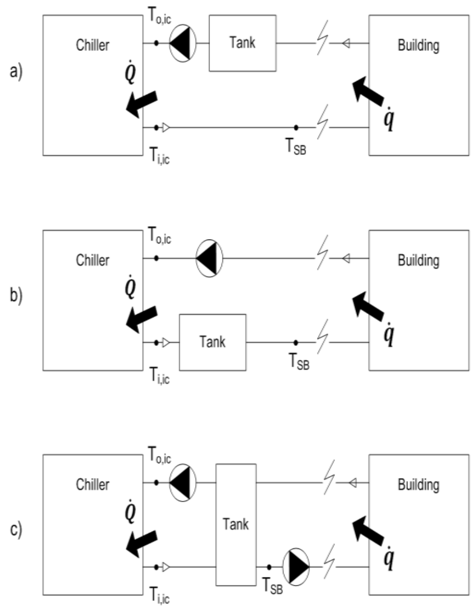

Figure 1: Different connection principles: a) tank return, b) tank supply, c) decoupling tank

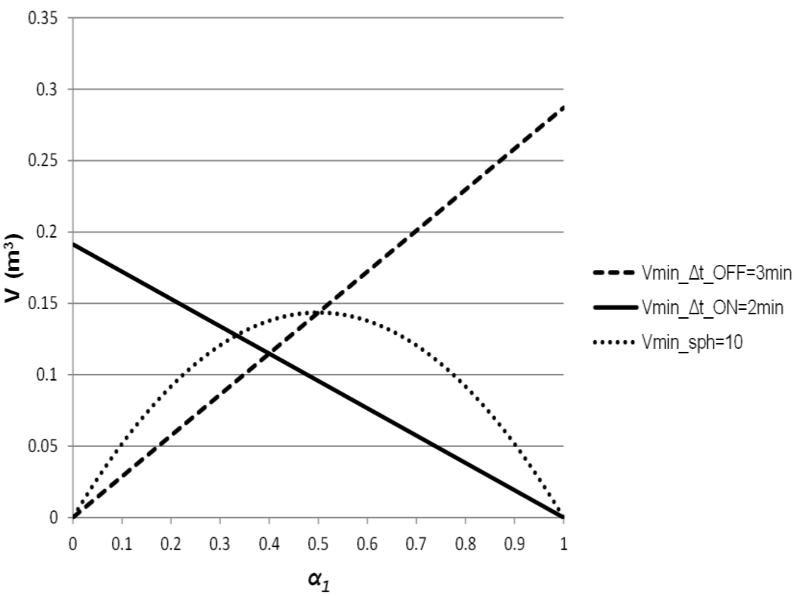

Figure 2: Example of required system volume in (m^3) as a function of the partial load ratio for a single compressor 20kW chiller