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**SIZE MATCHING CONSIDERATION DURING REPLACEMENT OF THE  
CONDENSING UNIT**

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**ABSTRACT**

Using a computer simulation this paper presents a few selection implications when the condensing unit of an existing R22 water-cooled chiller is to be replaced by a new unit that is originally sized for a different refrigerant. The selection of a new unit is assumed to be based purely on the information available in typical product catalogues. The performance of the upgraded (or retrofitted) system and changes to charge requirement of the chiller are discussed. The results provide an insight as how to select a suitable unit to deliver the desirable capacity in retrofitting circumstances. Issues related to selection of lubricant and material compatibility etc. are not considered for simplicity. It appears that the displacement rate of the replacing compressor has a noticeable influence on the performance of the upgraded system.

**NOMENCLATURE**

<b>Notations</b>		<b>Subscripts</b>	
A	Area (m <sup>2</sup> )	bubble	Bubble point
HTF	Heat transfer fluid	con	Condenser (or condensing)
CU	Condensing unit	cmp	compressor
Q <sub>e</sub>	Cooling capacity (kW)	dew	Dew point
RAC	Refrigeration and air-conditioning	evp	Evaporator (or evaporating)
T	Temperature (°C)	htf	Heat Transfer Fluid
W	power consumption (kW)	mid	mid-point
ΔM	Change of refrigerant charge (kg)	suc	Suction side of compressor
V	Displacement rate (m <sup>3</sup> /h)		

**INTRODUCTION**

When designing and building RAC systems of relatively small capacities, it is a common practice to obtain directly an off-the-shelf condensing unit and match (or balance) it to an air or a liquid cooler. This approach is convenient due to the fact that pre-fabricated condensing units usually come complete with the necessary piping in place among compressor, condenser, oil separator and accumulator, etc. The system balancing task is also simplified, as there is no need to balance the individual components one by one (i.e. compressor and condenser).

However when there is a need to replace the condensing unit of an existing system, due to, say, motor burn out, it may not be always possible to obtain exactly the identical unit due to product unavailability. If a different unit is selected, the changes on system capacity and operating parameters must be evaluated carefully, since the compressor and/or condenser sizes could be different, and their size-relationship are already fixed for a given unit. In cases where the new unit is originally sized for a different refrigerant, in particular a mixture refrigerant, further care must be exercised. It must be borne in mind that information such as condenser heat transfer area and compressor displacement rate are usually not given in the catalogue, though the latter may be found in some catalogues.

Initial selection of pre-fabricated condensing units from typical catalogues is normally based on cooling capacity at desired condensing and evaporating temperatures. Two possible scenarios exist. First, if a unit is selected from a catalogue in which the units are developed for using the same refrigerant, the changes in capacity and operating temperatures can be determined by the conventional graphical balancing techniques. In cases where the capacities are quoted under different specified conditions of suction superheat, motor speed etc., correction factors can easily be applied.

Second, when selecting units developed for a different refrigerant, it is less straightforward to predict the consequence due to a greater possibility in mismatching the condenser heat transfer area and the compressor displacement rate. This can also be interpreted as respective changes in the condenser and compressor characteristics caused by different refrigerant properties. The standard balancing procedure is therefore not applicable. The selection process could further be clouded by the possible use of different reference temperatures (dew point, bubble, etc.) in component's rating in case of mixture refrigerants. In these instances, attention should also be focused on the compressor motor power consumption, as there is a risk of overloading the motor. Depending on the size of the condenser, refrigerant charge of the system may need adjusted.

To restore the system cooling capacity as desired in the latter case, knowledge on how the performance characteristics of the selected unit change with the refrigerant already in the system is necessary. This information provide an insight as to how the overall system performance varies, that in turn helps identify a suitable unit from the catalogue. However, for inexperienced engineers, the changes of performance and charge requirement when coupled with existing system are relatively difficult to state with a given accuracy at the catalogue selection stage. Using a computer simulation, this paper looks at the implications of selecting condensing units from catalogues, which are either produced for R22 (the original refrigerant) or a different refrigerant (including mixtures). The performance of retrofitted or upgraded system is discussed in the context of variation of cooling capacity, refrigerant charge and compressor power consumption. Though the analysis is based on simplified water-to-water system, the findings are generally applicable to air-cooled units as well. The paper however excludes the consideration in relation to oil and material compatibility.

## SYSTEM SIMULATION APPROACH

### Existing R22 system

The system considered in this study, as shown in Fig. 1, is a simplified single stage vapour compression water-to-water cooling (chiller) system. It communicates with the space being cooled (load) via a water loop. A 7.5 kW designed cooling load is met with a 5 °C temperature drop of the HTF entering the evaporator at 14 °C, at a COP of 6.9. Some of the design data of the system are given in Table 1.

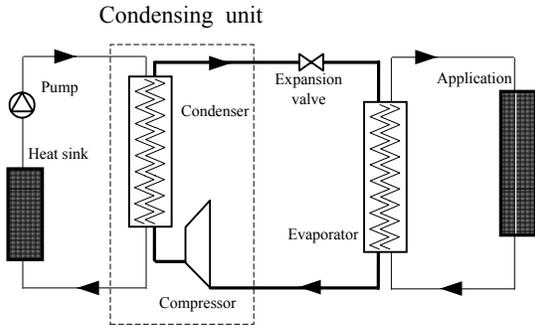


Fig. 1 Schematic of R22 cooling (chiller) system

Table 1 Design parameters of R22 system

Parameter	Condenser	Evaporator
Temperature (°C)	40	3
Pressure (kPa,)	1534	548
Heat Transfer Area /(m <sup>2</sup> )	0.37	0.54
Water in and out temperature/(°C)	27 34	14 9
Compressor displacement rate/(m <sup>3</sup> /h)	7.14	
System charge R22 (kg)	3.76 kg	

### Component models

Both condenser and evaporator use a phase-wise calculation approach [1,2]. The phase-changing section is divided into a number of small elements. Evaluation of thermal quantities in each element are based on local pressure, temperature and refrigerant state. The required refrigerant properties are estimated by internally calling property subroutine of NIST database 23 (REFPROP) [6]. Local values of heat transfer coefficients (HTC) and pressure drop are estimated using selected correlations from published literature [3,4,5].

When sizing the heat exchangers, stream to stream temperature differences are varied within respective model to obtain the required condensing and evaporating temperatures. HTF flow rates are set to achieve a specified temperature change of HTF under a given load. For the expansion valve, isenthalpic expansion is assumed and the superheat at evaporator exit is specified.

Compression is represented by a polytropic process. A reciprocating compressor is assumed in sizing the compressor displacement rate, which is necessary to obtain refrigerant mass flow rate during iterations at different evaporator pressures.

The quantity of the refrigerant in the system or in a given heat exchanger is estimated by separating the condenser and the evaporator into sections according to refrigerant phases. Two-phase quantity is derived using Humark void fraction model [7].

Following a similar style of data presentation in a typical catalogue, specification and performance data of different condensing units (models) with each selected refrigerant are

obtained using the simulation models. Condensing and evaporating temperatures and capacities provided in subsequent tables are shown to represent the particular section of a catalogue where it is likely to find appropriate units. The sizes of compressor and condenser, charge, compressor power consumption and delivered cooling capacity when retrofitted with a selected unit are discussed in comparison to the original chiller performance. The information could help to explain how the catalogue data should be interpreted to obtain a suitable unit to provide the required performance, in relation to the refrigerant used.

## DIFFERENT CONDENSING UNITS AND RETROFITTED SYSTEM PERFORMANCE

### R22 condensing units

Simulated catalogue data for a few R22 condensing units at the required condensing temperature of 40 °C, closer to the desired capacity of 7.5 kW at different  $T_{evp}$ , are given in the Table 2. The study assumes that a condensing unit identical to the one being replaced is not available. The data ( $A_{con}$  and  $V_{cmp}$ ) in the columns 5 and 6 are included to provide an understanding on how the component sizes vary across different models in a catalogue.

Table 2 Catalogue data for R22 condensing unit and performance of the retrofitted system

<b>R22 Cooling capacity (kW)</b>				<b>Components</b>		<b>Coupled to existing chiller</b>				
Unit	Evaporating temperature (°C)			$A_{con}$ (m <sup>2</sup> )	$V_{cmp}$ (m <sup>3</sup> /h)	$Q_e$ (kW)	$W_{cmp}$ (kW)	Charge (kg)	$T_{evp}$ (°C)	$T_{htf\ out}$ (°C)
	10	7.5	5							
CU1	<b>7.46</b>	6.88	6.40	0.281	5.70	6.23	0.88	3.18	4.0	10.01
CU2	8.11	<b>7.53</b>	7.00	0.328	6.23	6.69	0.97	3.39	3.5	9.58
CU3	8.75	8.13	<b>7.51</b>	0.368	6.69	7.12	1.03	3.67	3.2	9.21

Specified conditions: water flow rate = 0.3 kg/s, condensing temperature = 40 °C Suction gas temperature = 20 °C, 5 K subcooling, motor speed = 1450 rpm

A condensing unit (CU1, CU2 or CU3) offering almost similar capacity at the same  $T_{con}$ , but at a higher  $T_{evp}$  than 3 °C, would generally imply that it will have a smaller condenser and compressor than that of the design case. Selection of such a unit would result in a lower system capacity, ranging from 6.23 to 7.12, when compared to the original design. As expected the best choice among the three options is CU3 which gives about 5% less capacity than the desired value. To achieve the original system capacity, a unit offering approximately 7.5 kW at a combination of higher  $T_{con}$  and  $T_{evp}$  could be selected. In other words if a condensing unit with 7.5 kW at  $T_{con} = 45^\circ\text{C}$  and  $T_{evp} = 5^\circ\text{C}$  was available, a better matched system capacity could be achieved.

### Condensing units from R134a catalogues

Selected simulated catalogue data for R134a units with approximately 7.5 kW at 4 different evaporating temperatures are given in Table 3. The table also presents the comparisons of charge requirement, system cooling capacity and power consumption when the system runs on either R22 or R134a. The sizes of compressors show distinct increase over the original R22 unit

in table 1 (due to relatively lower vapour density and latent heat of vaporisation of R134a). All the first four R134a condensing units consume significantly higher compressor power and deliver different cooling capacities when the system runs on R22. The resulting evaporating temperatures will be slightly below 3°C for CU1 and around 0.5°C for the system using CU4.

Observations suggest that if a unit with around 7.5 kW capacity at a  $T_{evp}$  close to 3 °C (i.e. CU3) was chosen, the system capacity and compressor power consumption would increase by around 30% and 70% respectively. In general if the original cooling duty was to be attained, the selection could aim at a unit with similar capacity but with a correspondingly higher  $T_{evp}$ , i.e. by choosing CU1 to provide 7.7kW of cooling. However such a selection approach is not recommended, as it would mean a possible motor overload and a significant drop in system efficiency or COP. Based on results, it can be suggested that a unit from the same catalogue (CU5) with a smaller specified capacity (say, around 5 kW) at 5 °C evaporating temperature can be chosen to restore the original system cooling capacity. The resulting evaporating temperature will be slightly higher than the design  $T_{evp}$  of 3 °C, and the compressor power consumption and charge (R22) requirement are relatively lower.

Table 3 Comparison of catalogue data and component sizes of R134a condensing units, and performance when coupled with existing system

R134a Cond. unit	Catalogue data Capacity (kW)				% Change compared to original R22 system						
	Evaporating temperature				Component size		Performance and refrigerant charge				
	12.5 (°C)	10 (°C)	5 (°C)	0 (°C)			with R22			with R134a	
				$A_{con}$ (%)	$V_{cmp}$ (%)	$Q_e$ (%)	$W_{cmp}$ (%)	$\Delta M$ (%)	$Q_e$ (%)	$W_{cmp}$ (%)	
CU1	7.49	-	-	-	-37	12	<b>3</b>	<b>41</b>	<b>-32</b>	-23	-27
CU2	-	7.72	-	-	-26	23	13	50	-23	-16	-19
CU3	-	-	7.50	-	-4	44	29	71	-10	<b>-4</b>	<b>-7</b>
CU4	-	-	-	7.52	22	67	47	91	6	8	7
CU5	-	-	4.90	-	-53	-8	-2	-9	-48	-	-

Specified conditions: water flow rate = 0.3 kg/s, condensing temperature = 40 °C,  
Suction gas temperature 20 °C, 5 K subcooling, motor speed = 1450 rpm

Care must also be exercised, however, to check if the capacities are quoted under the same test conditions for suction and subcool temperatures, etc. For example when selected at 5°C evaporating and 40 °C condensing from a catalogue that specifies suction temperature at 10°C, instead of 20 °C, a unit with approximately 7.5 kW capacity would have about 5 % larger compressor than CU3 in the table 3. This unit would deliver about 35% (instead of 29%) higher cooling capacity. The above exercise is purely used to demonstrate the influence of thermal properties on component matching. However one would not normally use R22 in R134a components due to its appreciably higher operating pressures than R134a, unless the original system was designed for a higher pressure rating. On the other hand if the retrofitted system is to be charged with R134a, it appears that a selection based on catalogue specified conditions close to desired values is likely to restore the original system performance, i.e. by selecting CU3, with only a 4% drop in capacity.

## Condensing units from catalogues for mixture refrigerants

Unlike with pure refrigerants, a condensing (or an evaporating) process of a mixture can be specified based on several possible reference temperatures; for example dew or bubble point or mid-point of the two. This leads to ambiguity in the choice of the temperatures in relation to selecting components from standard catalogues.

When rating compressors for mixture refrigerants, the pressure corresponding to dew or mid-point temperature can be used [8]. With each of these pressure/temperature definitions, for a given compressor, there will be a difference in capacity between what is given in the catalogue and what is delivered under actual operating conditions. However, selecting a compressor based on the mid point generally results in a smaller discrepancy as the actual evaporating conditions take place at a pressure close to that corresponds to mid-point pressure [8].

In catalogues for mixture condensing units, information about the reference temperature for the presented data is not normally specified clearly. When using such catalogue, two possibilities of interpretations exist:

1. The specified temperatures refer to dew point as the  $T_{con}$  and  $T_{evp}$ .
2. The specified temperatures,  $T_{con}$  and  $T_{evp}$ , refer to the  $T_{mid}$  values (i.e. approximately equal to the average of  $T_{dew}$  and  $T_{bubble}$ ).

Changes of component sizes and system parameters for two condensing units selected from two separate R407C catalogues are shown in Table 4. One catalogue is assumed to have its data presented based on the first interpretation and the other based on the second. In other words, 5 °C for CU1 is interpreted as dew but as mid-point for CU2. Since the temperature glide is relatively smaller for R407C, differences in the condenser and the compressor sizes between the two units are well within 10%. With each unit, the capacity of the original system is almost restored. The evaporating temperatures of retrofitted system with either unit remain relatively close to the original design temperature. The increases in charge requirement are also similar for the two refrigerants. Similar performance with either refrigerant is owing to the fact that R407C is a look-alike (especially almost similar vapour densities and latent heat) replacement for R22.

Table 4 Comparison of catalogue data and component sizes of R407C condensing units, and performance when coupled with existing system

R407C Cond. Unit	Catalogue data		% Change compared to original R22 system						
			Component size		Performance and refrigerant charge				
	$T_{evp}$ = 5 °C	$Q_e$ (kW)	$A_{con}$ (%)	$V_{cmp}$ (%)	with R22		with R407C		
				$Q_e$ (%)	$W_{cmp}$ (%)	$\Delta M$ (%)	$Q_e$ (%)	$W_{cmp}$ (%)	
CU1	$T_{dew}$	7.42	31	2	4	-3	20	4	5
CU2	$T_{mid}$	7.34	40	-3	-1	-11	27	0	-3

Specified conditions: water flow rate = 0.3 kg/s, Suction gas temperature 20 °C, 5 K subcooling, motor speed = 1450 rpm

However larger differences in the component sizes are expected, under the two interpretations, for units using mixtures with large temperature glides. For instance, if a condensing unit is selected from a catalogue for a R142b+R143a mixture, it must be ascertained that the reference temperatures are properly defined and understood, As demonstrated in Table 5, the condenser and compressor sizes in CU1 and CU2 can be different by a large margin. The resulting changes in performance and charge requirement when run on R22 also depend on the difference in thermal-physical properties between R22 and the mixture, in addition to the glide influence on component sizes. To achieve the original capacity for this particular mixture, one should select a smaller catalogue capacity at similar  $T_{\text{evp}}$  and  $T_{\text{con}}$ .

Table 5 Comparison of catalogue data and component sizes of R412b/R143a mixture condensing units, and performance when coupled with existing system

R143a/ R142b Cond. Unit	Catalogue data		% Change compared to original R22 system						
			Component size		Performance and refrigerant charge				
	$T_{\text{evp}}$ = 5 °C	$Q_e$ (kW)	$A_{\text{con}}$ (%)	$V_{\text{cmp}}$ (%)	with R22		with mixture		
				$Q_e$ (%)	$W_{\text{cmp}}$ (%)	$\Delta M$ (%)	$Q_e$ (%)	$W_{\text{cmp}}$ (%)	
CU1	$T_{\text{dew}}$	7.46	7	68	54	102	1	12	31
CU2	$T_{\text{mid}}$	7.38	22	45	43	63	12	1	4

Specified conditions: water flow rate = 0.3 kg/s, Suction gas temperature 20 °C, 5 K subcooling, motor speed = 1450 rpm. A R142b+R143a (50/50 %by mass) mixture has a glide of 14 °C at atm. pressure.

## CONCLUSION

Implication of selecting condensing units from catalogues for different refrigerants, to replace a R22 unit, was investigated in the context of component sizes, performance and refrigerant charge of the retrofitted system. The selection of a unit is to be purely based on the information available in standard catalogues. Based on the result, the following observations can be made:

- when selected from a catalogue for a different pure or mixture refrigerant, component sizes of condensing units are generally different to those being replaced.
- size of the compressor largely decides the change of cooling capacity of retrofitted system when run without changing the original refrigerant whereas, as expected, size difference of the condensers mainly influences the refrigerant charge.
- selection of a R134a unit at catalogue capacity and temperatures close to the desired values increases both the system capacity and risk of compressor motor failure. Such selection approach is on the other hand appropriate for the R407C replacing unit.
- selecting a unit from a catalogue for a mixture with significant glides needs to consider the effects of thermal-physical properties and the definition of the specified temperatures.

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