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SIMULATION OF POTENTIAL REFRIGERANTS FOR RETROFIT REPLACEMENT

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ABSTRACT

This paper shows two approaches of refrigerant comparison for retrofit replacement. One is the screening based on refrigerant thermo-physical properties and cycle performance analysis, another method is the full simulation. The impact of component sizes on the refrigerant temperatures and performance is included in the full simulation while it is ignored in the former analysis. The methods are exemplified by comparing some common refrigerants, including R32, R410A, R125, R1270, R22, R407C, R290, R134a, R600a and R600.

Keywords: refrigerant, screening, full simulation, retrofit, replacement.

Nomenclature

A area [m^2]
c_p isobaric specific heat [kJ·kg^{-1}·K^{-1}]
C heat capacity = mass flow rate×isobaric specific heat [kW·K^{-1}],
COP coefficient of performance ($= Q_{cool} / W_{com}$) [---]
GWP global warming potential
h specific enthalpy [kJ·kg^{-1}]
$HCFC$ hydro chlorofluorocarbon
HX heat exchanger
HTC heat transfer coefficient [W·m^{-2}K^{-1}]
HTF heat transfer fluid
L length [m]
$\dot{m}$ mass flow rate [kg·s^{-1}]
NBP normal boiling point [°C]
NTU number of transfer unit [---]
ODP ozone depleting potential [---]
P pressure [kPa]
Q capacity [kW]
SC degree of sub-cooling [°C or K]
SH degree of superheating (for evaporator), or degree of de-superheating (for condenser) [°C or K]
T temperature [°C or K]
$\dot{V}_{sw}$ compressor swept volume rate [m^3/s]
VRC volumetric refrigerating capacity or effect ($= \Delta h_{refrig} / v_{suc}$) [kJ/m^3]
W compressor work [kW]
Δ the difference
Δh enthalpy difference [kJ·kg^{-1}]
Δh_{refrig} refrigerating effect [kJ·kg^{-1}]
ΔT_w the difference between inlet and outlet HTF temperatures of the entire heat exchanger [°C]

Greek symbols

ε effectiveness
η efficiency [%]
ρ density [kg·m^{-3}]
v specific volume [m^3·kg^{-1}]

Subscripts

act actual
c, cond condenser or condensation or conduction
cool cooling
com compressor
dew dew (saturated vapour state)
dis discharge
e, evap evaporator, evaporation
exv expansion valve
in inlet
isen isentropic
min minimum
out outlet
r refrigerant
SC sub-cooling
SH superheating
suc suction
vol volumetric
w water or heat transfer fluid

INTRODUCTION

Among available refrigerants, R-22 has been widely used for many years. It possesses many desirable physical and thermodynamic properties and can be employed in a wide range of applications and temperatures with good system performance. It is also safe in terms of toxicity and flammability. Nevertheless, in response to Montreal Protocol [1], R22, as the last remaining ozone depleting HCFC, will face the eventual phase-out in probably less than 5 to 10 years time [1]. Many alternative refrigerants have been developed to replace R22 as well as many of those already phased out. The choice of alternative refrigerants is vast and it is not always easy to make the appropriate decision, though many of the new refrigerants are expected to deliver the same or even better energy performance than those being phased out.

Several scenarios of refrigerant substitution can be adopted; these are “drop-in”, “retrofit”, and “new” systems. “Drop-in” - where the old refrigerant is taken out and the system charged with the alternative refrigerant and

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occasionally with some minor adjustments to the control settings, "retrofit" - where the old refrigerant is replaced with an alternative refrigerant often accompanied by oil and material changes due to compatibility issues, and "new system" - replacement of old systems with new ones designed specifically for the alternative refrigerant [2].

The objective of this paper is to demonstrate the differences of the outcome from the cycle analysis and the simulation technique for retrofit refrigerant comparison; their limitations are also discussed. The methods are exemplified by comparing some common refrigerants, e.g. R32, R410A, R125, R1270, R22, R407C, R290, R134a, R600a and R600 for the cycle analysis. For the full simulation, R417A, 422D, R427A are included as they are recommended as retrofit choices by refrigerant companies while excluding R600 since its properties are rather similar to R600a.

Screening based on thermo-physical properties and cycle performance analysis

Many factors need to be considered when deciding upon whether certain alternatives can be suitably used as replacement refrigerants [3], [4], [5]. Apart from assessing the environmental factors, i.e., *ODP* and *GWP*, as well as the safety and material compatibility issues, the preliminary screening of potential replacements for given application temperatures can be based on the T_{crit} and *NBP* of the refrigerants. Further evaluation can employ cycle analysis, examining factors such as P_{cond}, T_{dis}, *COP* and *VRC*, and matching them with the corresponding values of the replaced refrigerant for the specified $T_{r,dew,evap}$, $T_{r,dew,cond}$, η_{isen}, SH_{evap} and SC_{cond}.

In general, the replacement refrigerants should have an appropriate T_{crit} and *NBP* for a given application temperature [6], i.e., the refrigerant should have T_{crit} that is higher than $T_{r,dew,cond}$ and *NBP* that is lower than $T_{r,dew,evap}$. In addition, to replace refrigerant in an existing system, the *VRC* should be close to provide similar capacity [7], and P_{cond} for a given ambient temperature should be no more than that of the original refrigerant. In addition, a higher *COP* is preferred [7], [8].

Results and discussions of cycle analysis

Table-1 shows the refrigerant properties and calculated cycle operation parameters at specified conditions for initial screening purpose (ordered from high to low *VRC*) [9]. As explained in McLinden and Didion [6], T_{crit} should be a trade-off between *COP* and capacity. R125 has a very low T_{crit} resulting in a poor *COP*, though it offers a much higher capacity than R22 due to its larger ρ_{suc} value despite having a much lower Δh_{refrig} than R22.

Table-1. Refrigerant properties and performance calculated at given conditions [9].

Refrigerant	*GWP*	T_{crit}	*NBP*	ρ_{suc}	Δh_{refrig}	*VRC*	P_{cond}	T_{dis}	*COP*
		(°C)	(°C)	kg/m^3	kJ/kg	kJ/m^3	kPa	(°C)	---
R32	550	78.1	-51.7	22.5	257	5790	2478	89.8	4.08
R410A	2000	70.4*	-52.7*	31.1	171	5325	2419	71.7	4.01
R125	3400	66.0	-48.1	43.0	91	3908	2008	52.7	3.81
R1270	20	92.4	-47.7	12.6	297	3745	1652	64	4.19
R22	1700	96.1	-40.8	21.8	166	3615	1534	75.7	4.26
R407C	1700	86.0*	-43.7*	20.3	171	3464	1541	66.9	4.17
R290	20	96.7	-42.1	10.6	292	3095	1370	58.2	4.22
R134a	1300	101.1	-26.1	14.9	155	2310	1017	59.1	4.29
R600a	20	134.7	-11.7	4.4	281	1234	531	52.3	4.38
R600	20	152.0	-0.6	2.9	311	893	379	50.9	4.45

Note: 1) All refrigerants shown have zero *ODP* except R22 which has *ODP* = 0.034 [5]. 2) All properties are obtained from REFPROP7.0 except *GWP* are from [5]. 3) * *NBP* of mixtures are approximated from their compositions. 4) Conditions for cycle analysis are $T_{r,dew,evap}$ = 1.67°C, SH_{evap} = 5°C, $T_{r,dew,cond,}$ = 40°C, SC_{cond} = 5°C, η_{isen} = 70%.

On the other hand, R600 though has a higher T_{crit}, it is considered not a suitable retrofit choice for R22 system as it has a much lower *VRC* due to its very low suction density, and hence a lower capacity. Also from *NBP* point of view, R600 is not suitable retrofit replacement for sub-zero application temperatures, and so is R600a. P_{cond} of R410A and R32 are very much higher (by about 60%) than that of R22, suggesting it is not suitable for retrofitting R22 systems, constrained by the system pressure limit. R134a has a far too low *VRC* (36% lower than R22) suggesting it is unsuitable for retrofit unless the compressor size or speed is increased.

R407C can be used in retrofit, supported by the experimental work of Greco *et al*., [10], with both its *VRC* and *COP* slightly less, and P_{cond} a little higher, than that of R22. In addition, T_{dis} is another factor relating to the system reliability in terms of oil and refrigerant degradation [11], and heat transfer loss, thus a low T_{dis} is preferred.

Compared with R22, R1270 and R290 both have lower discharge temperatures, and putting aside the flammability issues, they could be considered as potential alternatives. As already pointed out, *VRC* is indeed the product of suction density and specific refrigerating effect. Therefore, though R1270 has a much lower suction density than R22, it also has a much higher Δh_{refrig}, resulting in a rather similar *VRC* and capacity as R22.

In summary, as suggested by many researchers T_{crit}, *VRC*, P_{cond} and *COP* at given temperatures ($T_{r,dew,evap}$ and $T_{r,dew,cond}$) can be used to screen for retrofit replacement. In initial screening, $T_{r,dew,evap}$, $T_{r,dew,cond}$, and η_{isen} are normally specified the same for all refrigerants being compared. However, in retrofit these parameters are likely to be different and they must be evaluated properly. The numerical approach implemented in this study, so called the full simulation, is shown in the following section.

The comparison of refrigerant performance for a given system using a full simulation

Full simulation method

The term "full simulation" adopted in this study refers to dividing the heat exchangers into small elements and grouping the elements into either single-phase or two-phase zone. For the evaporator, there are two-phase evaporation and single-phase superheating zones; for the condenser there are two-phase condensation, and single-phase sub-cooling and de-superheating zones. *NTU*-effectiveness method is applied for both the heat exchanger sizing and the capacity rating calculation. The system model for the full simulation consists of an evaporator, a condenser, a compressor and an expansion valve. The system characteristics and numerical procedure are described below.

Heat exchanger

A tube-in-tube counter-flow *HX* is employed for both the evaporator and the condenser. Refrigerant is in the inner tube, while the water, as the *HTF*, is in the annulus. A conventional numerical approach using element-by-element *NTU*-effectiveness [12] is adopted. Heat balance equations in an evaporator are shown as in Equations (1) to (3).

For refrigerant side:

$$Q_{evap} = \dot{m}_r \Delta h_r \tag{1}$$

For heat transfer between the refrigerant and the *HTF*:

$$Q_{evap} = \varepsilon C_{\min}\left(T_{w,in} - T_{r,in}\right) \tag{2}$$

For heat transfer fluid side:

$$Q_{evap} = \dot{m}_w c_{p,w} \Delta T_w \tag{3}$$

Similarly, these three heat balance equations also apply to the condenser heat rejection, using the corresponding properties and parameters in the condenser.

The methodology of Breber *et al*. (1980) (cited in [13]) is chosen for the two-phase flow regime prediction in the *HX*s, from which appropriate *HTC* correlations were chosen. Gungor and Winterton (1987) and Cavallini and Zecchin (1974) (cited in [13]) correlations are used, respectively for evaporation and condensation *HTC*.

Compressor

An open-typed reciprocating compressor is employed. It is characterised by using its isentropic and volumetric efficiencies to determine work input and mass flow rate, [14], respectively as in Equations (4) and (5).

$$W_{act} = \frac{\dot{m}_r \Delta h_{isen}}{\eta_{isen}} \tag{4}$$

$$\dot{m}_r = \rho_{suc} \dot{V}_{sw} \eta_{vol} \tag{5}$$

Expansion valve

The expansion process is modelled as an isenthalpic process and the degree of superheat at the evaporator outlet is assumed fixed at all times [14] as in Equations (6) and (7), respectively.

$$h_{exv,in} = h_{exv,out} \tag{6}$$

$$SH_{evap} = \text{constant} \tag{7}$$

Assumptions

The assumptions adopted in the full simulation are as following.

a) Pressure drops in the system are ignored.
b) The SH_{evap} and SC_{cond} are kept constant for all refrigerants.
c) The refrigerant charge quantity is always enough to provide the specified SC_{cond}.
d) The inlet temperature and mass flow rate of *HTF* through the *HXs* are kept constant before and after the retrofit.
e) Isentropic and volumetric efficiencies are specified the same for all refrigerants. Though, these parameters can influence the ranking of refrigerant performance comparison [9].

Results and discussions of the full simulation

The design conditions are shown in Table-2. The system performance relative to R22 and refrigerant conditions for different refrigerants are shown respectively in Figure-1 and Table-3. The system was originally sized for R22 at a cooling capacity (Q_{cool} or Q_{evap}) of 8 kW (*COP* = 4.26) at the design conditions (shown in Table-2), providing an A_{evap} = 1.10 m^2, A_{cond} = 0.85 m^2 and a compressor swept flow rate = 0.002454 m^3/s (or 8.83 m^3/h). It is noted that A_{cond} is smaller than A_{evap} due to smaller ΔT_w and higher *HTF* mass flow rate in the condenser. Refrigerant safety issues and pressure limit are

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not considered, and the compressor motor rating is assumed sufficient to cope with additional power requirement, if needed. In Figure-1, the refrigerants are ordered from low (on the left) to high cooling capacity and in Table-3, they are ordered from low to high evaporating temperatures.

Table-2. Design conditions for component sizing.

Q_{cool} (kW)	8	$T_{w,in,evap}$ (°C)	12.2
$T_{r,dew,evap}$ (°C)	1.7	$\Delta T_{w,evap}$ (°C)	5.5
$T_{r,dew,cond}$ (°C)	40	$T_{w,in,cond}$ (°C)	29
SH_{evap} (°C)	5	$\Delta T_{w,cond}$ (°C)	3
SC_{cond} (°C)	5	inner tube diameter (mm)	17
η_{isen} (%)	70	outer (annulus) tube diameter (mm)	34
η_{vol} (%)	90		

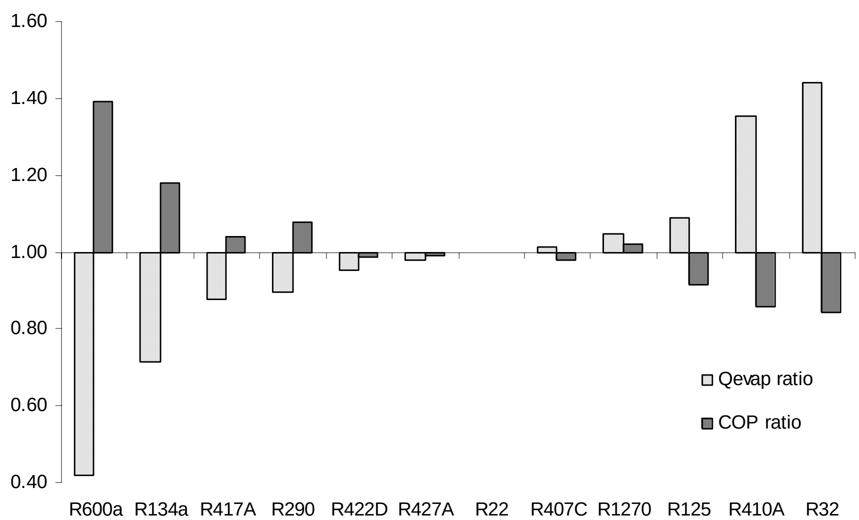

Figure-1. The ratios of cooling capacity and *COP* of different refrigerants in retrofit system normalised to those of R22 [9].

The results clearly show that when retrofit an existing R22 system to another refrigerant, both Q_{cool} and *COP* change together with the refrigerant temperatures. Refrigerants that provide too low Q_{cool} such as R600a and R134a are not suitable for retrofit, unless the compressor is changed (to a larger size or a higher speed). Refrigerants with too high Q_{cool} such as R410A and R32 are not appropriate either, because much more additional work input is required to drive the much higher capacity and the existing motor may have insufficient power rating. Therefore, the refrigerants that provide within ±10% of the baseline capacity in this figure may be considered as potential candidates for retrofit.

Table-3. Replacement refrigerant dew point temperatures and evaporator outlet water temperatures [9].

	$T_{r,dew,evap}$ (°C)	$T_{r,dew,cond}$ (°C)	$T_{w,evap,out}$ (°C)
R32	-1.2	41.2	4.3
R410A	-0.5	40.6	4.8
R125	1.4	39.2	6.2
R22 (baseline)	1.7	40.0	6.7
R1270	1.7	39.0	6.5
R290	2.7	38.6	7.3
R422D	3.5	40.4	7.0
R407C	3.9	42.3	6.7
R427A	4.0	41.9	6.8
R417A	4.1	40.3	7.4
R134a	4.1	37.9	8.3
R600a	6.2	36.2	9.9

The difference of Q_{cool} in Figure-1 can be explained by their differences in *VRC* and refrigerant temperatures. Since the *VRC* varies with refrigerant temperatures, the results can be used to illustrate it is not appropriate to use the *VRC* at the same temperatures to evaluate the retrofit capacity. The retrofit capacity obtained for R290 is 10% smaller than R22 (Figure-1), though at the same temperatures (as previously seen in Table-1), the *VRC* of R290 is 14% smaller than R22. Likewise, the retrofit capacity of R407C is about 1% higher than R22 (Figure-1), despite at the same refrigerant temperature, *VRC* of R407C is in fact 4% lower than R22 (Table-1). The increase of capacities for R290 and R407C compared to those at fixed refrigerant temperatures is attributed to the fact that both refrigerants experience an increase in $T_{r,dew,evap}$ (Table-3) following a retrofit. The *HTF* outlet temperature from the evaporator (shown in the last column of Table 3) can be regarded as the reciprocal to the cooling capacities obtained, i.e. the higher cooling capacity, the lower *HTF* outlet temperature. This implies that the desired *HTF* temperature cannot be attained when the capacity does not match with the original one; either too low or too high *HTF* temperature could be experienced.

Experimental results from published literature were used to assure the simulation trends are correctly predicted. Review paper of Granryd [15] indicates that when retrofit R22 systems to R290, a lower cooling capacity and an improved *COP* than R22 were observed; same trends were obtained by the current simulation. Park *et al*., [16] compared R290 and R1270 with R22 experimentally in the same system at the approximately the same refrigerant temperatures. It was found that *COP* of R290 and R1270 are higher and lower (though slightly), respectively than that of R22; and cooling capacities for R290 and R1270 are lower and higher, respectively than that of R22. These again in general agree well with the simulation except that the simulation only predicts a marginally higher *COP* for R1270 than R22. The experimental results of Devotta *et al*., [17] showed that both cooling capacity and *COP* of R407C are lower than that of R22. However, the current simulation gives a slightly higher cooling capacity for R407C than R22. Inaccuracy in the simulation, the use of a different reference condition/temperature and assumption of having the compressor efficiency fixed could easily result in these small discrepancies.

The retrofit temperatures predicted by the simulation also agree well with published experimental and other simulated results. A higher $T_{r,dew,evap}$ of R290 compared to R22 obtained in a drop-in simulation of Domanski and Didion [18] and Devotta *et al*., [19] supports the current prediction. A higher $T_{r,dew,evap}$ for R407C was also observed by Devotta *et al*., [17]. A lower predicted $T_{r,dew,cond}$ of R290 also agrees well with Hammad and Tarawnah [20].

To summarize, in general the full simulation results agree well with the experimental results. Nevertheless, it is worth to emphasize that in this simulation, the compressor efficiencies are assumed the same for all refrigerants, which are unlikely to happen in practice. In addition, pressure drops are not considered in the simulation, while they are inherently included in the experimental results. Hence, some discrepancies are expected.

CONCLUSIONS

a) T_{crit}, *VRC*, P_{cond} and *COP* at given temperatures ($T_{r,dew,evap}$ and $T_{r,dew,cond}$) can be used to screen for retrofit replacement. In initial screening, $T_{r,dew,evap}$, $T_{r,dew,cond}$, and η_{isen} are normally specified the same for all refrigerants compared. However, in retrofit those parameters are likely to be different.

b) Compared with conventional cycle analysis which can only be performed under fixed temperatures, the full simulation can generate the retrofit temperatures so the results provide a more realistic representation of the practical situation.
c) In retrofit, the same original performance and/or capacity may not be attained due to the use of the existing heat exchangers and compressor, unless the two refrigerants have very similar properties.

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