Effects of refrigerant properties on refrigerant performance comparison: A review

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The effects of thermo-physical properties and derived parameters of refrigerants on system performance comparison are reviewed here. The properties reviewed are density, viscosity, thermal conductivity, molar heat capacity, critical temperatures, specific refrigerating effect and molecular weight. It was noted that to obtain a high COP, combinations of high values of latent heat, liquid thermal conductivity and vapour density and low values of liquid viscosities and molecular weight are required. Critical temperature and vapour specific heat are important properties when considering trade-offs between capacity and COP. The oil existance in the system could reduce or increase heat transfer or/and pressure drop comparing to those ignore the presence of oil depending on the amount of oil and solubility.

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 right 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 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].

For new systems, components can be designed and sized to suit individual refrigerants to achieve the required system capacities at specified refrigerant temperatures [3]. For examples, refrigerants with relatively poorer heat transfer characteristics will require a larger heat exchanger area [4] or better designed heat exchangers [5],[6] to achieve the same capacity. On the other hand, refrigerants with a high volumetric refrigerating effect (or volumetric refrigerating capacity, kJ/m3) can employ smaller compressors [7]. In such cases, the capital cost per unit capacity and the COP will probably be one of the main interests for the comparison. In both the drop-in and retrofit scenarios, 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. For example, when retrofitting a R22 system to R407C or R290, lower cooling capacities were obtained [8], [9], [10] due to their lower volumetric refrigerating effects.

This paper is to review relevant properties/parameters for the refrigerant performance comparison. First, properties for refrigerant only, including density, viscosity, thermal conductivity, molar heat capacity, critical temperatures, specific refrigerating effect and molecular weight, are reviewed. Derived parameters such as volumetric refrigerating capacity, heat transfer coefficient and pressure drop are reviewed in the following section. Finally, the refrigerant-oil properties/parameters are reviewed.

Effects of key refrigerant properties on system performance

Density or Specific volume

Density (ρ) or specific volume ($v = 1/\rho$) of a refrigerant influences pressure drop throughout the refrigerant circuit as well as the compressor capacity. The lower the vapour density, the larger will be the pressure drop in the evaporator and the condenser for a given mass flux, and thus for instance, R290 has a larger pressure drop compared to R404A, R410A and

R22 [11]. For a given compressor size and speed, a lower vapour density will also result in a lower capacity. As shown in Hwang and Radermacher [12], the saturated vapour density of R410A is approximately 40% higher than that of R22 (and R410A's latent heat is also 7% higher), for the same compressor, it will provide much higher capacity than R22. They needed to use a smaller compressor for R410A to obtain similar capacity as R22.

Viscosity

Higher values of liquid viscosity increase pressure drop in both evaporators and condensers [13]. As a result, suction pressure at the compressor inlet decreases, discharge pressure increases and the mass flow rate of refrigerant also decreases, followed by a reduction in system capacity. The compressor work (kW) also decreases, though at a slower rate than capacity. Higher liquid viscosities also give rise to reduced HTCs. Hydrocarbon refrigerants have lower liquid viscosities than R22, hence higher HTCs in the evaporator and condenser [14].

Thermal conductivity

In general, a higher liquid thermal conductivity would give rise to a higher HTC. With an equal mass flux or an equal heat flux, evaporation HTC of R290, R600 and R600/R290 are higher when compared to R134a as these refrigerants have higher thermal conductivities than R134a [15].

Molar heat capacity or Specific heat

The magnitude of the molar heat capacity at constant volume has a strong influence on the slope of the saturated liquid and vapour lines as shown on the T-s diagram, Figure 1 [16]. The slope of the saturated vapour line can be positive or negative. In general, refrigerants with more complex molecules are likely to have a higher molar heat capacity, and more likely that their saturated vapour line will bend to give a negative slope at the lower pressure/temperature and a positive slope at the higher pressure. When the heat capacity is too low, the compression work will be high with increased discharge temperature and reduced thermal efficiency. On the other hand, for a high heat capacity, wet compression may be encountered [17]. Therefore, a trade-off in the value of the heat capacity is needed.

Critical temperature

In general, as the critical temperature of a refrigerant increases the volumetric refrigerating capacity (refrigerating effect per specific volume, kJ/m3) decreases; this is due to the fact that lower vapour densities (at a given evaporating temperature) are usually associated with refrigerants with a high critical temperature [16], [17]. The implication is that

for a given compressor (speed and size), the reduction of suction density will lead to a decrease in the mass flow rate and hence the cooling capacity. On the other hand, when design a new system, a refrigerant with a high Tcrit will require a larger compressor swept volume rate to provide the same capacity. The COP, however, increases for refrigerants with a high critical temperature, due to reduced flash gas losses. Therefore, a trade-off between high capacity and high efficiency has to be considered [16], [17].

Specific refrigerating effect

The specific refrigerating effect determines the mass flow rate of refrigerant required to produce a given capacity. The larger the specific refrigerating effect, the smaller the required mass flow rate. It, therefore, also determines the size of the components [18]. R290, having a higher specific refrigerating effect (or a high latent heat), thus requires a smaller size of the compressor than R12 for providing the same capacity [19].

Molecular weight

Large values of enthalpy of evaporation are found for substances with light molecules [18], and the energy losses across a compressor's valves are high when the molecules are heavy [20]. Therefore a low value of molecular weight is preferred for having high efficiency [21].

To sum up, in general, to obtain high COP, the following combination of properties is desirable, as shown in Table 1. Note that for critical temperature and vapour specific heat, a trade-off between capacity and COP should be considered.

Domanski and Didion [13] and Hogberg and Vamling [22] showed that each individual refrigerant property affects the system performance with different sensitivities or by different extents. As shown in Table 2, the normal boiling point, the critical temperature, the liquid thermal conductivity and viscosity, the vapour density and the evaporative HTC appeared to have the most impact on system performance. Evaporator pressure drop has only a slight influence whereas the vapour-phase thermal conductivity and viscosity, together with the condensation HTC and condenser pressure drop, were found to have the least impacts.

Effects of derived parameters on system performance

Volumetric (or volumic) refrigerating capacity ($VRC = \Delta h_{refrig}/v_{suc}$, kJ/m³)

In general, fluids with a higher vapour pressure would also have higher values of volumetric refrigerating capacity (VRC) and thus require a smaller compressor displacement rate [23], [24] to deliver the

similar capacity. To replace or retrofit an existing refrigerant without changing the compressor, VRC of alternative fluids should be similar to that of the replaced fluid [25]. However, whether the retrofit COP will increase or decrease depends on many other factors, such as the retrofit refrigerant temperatures, pressure ratio and compressor efficiencies.

Pressure ratio

In general, when pressure ratio increases, both isentropic and volumetric efficiencies of a reciprocating compressor decrease [26], [27], [28] influencing the COP and mass flow rate, respectively.

Heat transfer coefficient (HTC) and pressure drop

The results of refrigerant performance comparison when only considering the thermodynamics may be different from those when taking into account the heat transfer and pressure drop in the system. Spatz and Motta [11] showed that, the COPs of a system using R410A and R290, based on thermodynamic analysis, are slightly lower than that of R22 when all are operating under the same capacity.

However, when including heat transfer and pressure drop effects of the individual refrigerants in the heat exchangers, the reverse was observed. At the same mass flux, R410A has a higher HTC and a lower ΔP than R22. The lower impact of a given ΔP on saturation temperatures coupled with an improved heat transfer for R410A reduces the temperature lift between Tr,dew,cond and Tr,dew,evap, thus lowering the compressor power requirement. On the other hand, though at the same mass flux, the HTC of R290 is higher than R22, R290 also has a higher pressure drop. However at a given capacity, R290 has a much lower mass flux than R22, and hence its HTC and pressure drop are comparable to R22, resulting in the COP of R290 only marginally better than R22. Simulation work of Devotta et al. [9] also found the pressure drop of R290 in both HXs are lower than that of R22 when specifying a refrigerant mass flow rate of R290 half of that R22, which is approximately the value one would experience in drop-in tests.

Table 2 includes some examples of the influence of variations of HTC and pressure drop on capacity and COP based on the sensitivity studies carried by Domanski and Didion [13] and Hogberg and Vamling [22].

When comparing the HTC and pressure drop of different refrigerants, one could base the evaluation on the same mass fluxes [29] or the same specified heat fluxes [30]; the latter usually corresponds to same capacities. However, these two comparison approaches could lead to different possible conclusions regarding which refrigerant has a higher HTC or a lower pressure drop. Chang et al. [14] compared pure and mixture

hydrocarbon with R22 experimentally in a heat pump system. At the same mass fluxes, HTCs of the hydrocarbons are much greater than R22 whereas when compared under the same capacities, hydrocarbons show slightly lower values of HTC than R22 due to the fact the hydrocarbons have lower mass fluxes under the same capacity.

The oil effects

The oil effects on thermo-physical properties, heat transfer coefficients and pressure drops, and heat exchanger/system performance are reviewed as follow.

Effect of oil on thermo-physical properties, HTC and pressure drop

Lubricant oils have much higher boiling points (Tb) and molecular weights (MW) than most of the common refrigerants and therefore the Tb and MW of refrigerant-oil mixture will be higher than the refrigerants. The presence of miscible oil in a refrigerant increases its critical temperature, liquid viscosity, surface tension, specific heat and thermal conductivity; on the other hand, the oil presence reduces its critical pressure and liquid density [31], [32]. Oil is essentially non-volatile under typical refrigeration operating condition, and therefore it mainly affects the liquid-phase properties.

Thome [31] commented that vapour pressure of oil is extremely small compared to that of the refrigerant, resulting in practically very little oil entering the vapour phase with the refrigerant. Heat transfer and pressure drop characteristics of evaporators are very sensitive to the presence of oil [33], [34]. However, it is difficult to predict accurately the HTC due to oil's presence. In some cases, a small quantity of oil (e.g. 0.1 % oil by mass in a R410A/POE mixture) could improve HTC_{cond} and HTC_{evap} by up to 10% and 5%, respectively, while a larger oil mass fraction, say 5%, the HTC decreases more severely by 25% [35].

In general, a reduction in HTC and an increase in two-phase pressure drop, for both evaporator and condenser, could be expected when oil is present [31], [36], [37], [38]. In addition, the effect of oil on local HTC varies with vapour quality, for example a 5% by mass in R134a/oil, at a vapour quality (x) of 0.5, could reduce the HTCevap by 20%, and at x = 0.9, the HTCevap could drop by 80% [39], [31].

Effect of oil on heat exchanger/system performance

A heat exchanger performance simulated by Rajakphaksa [40] using R407C with an oil content of 0.5% − 2.5% of the refrigerant mass in the evaporator revealed a 2% reduction in overall heat transfer coefficient due to both the reduction in refrigerant mass flow rate and an about 25% increase in two-phase

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pressure drop. These caused about a 5% reduction in evaporator capacity.

Experimentally, Lottin et al. [41] found that typical content of 1 – 5% of oil by mass in R410A reduces the system COP by about 4 – 24%. Nevertheless, the effect of oil on the system performance is negligible when the lubricant does not exceed 0.5% of the total refrigerant weight in the system. For a particular refrigerant (e.g. R407C) using different oils with the same viscosity grade, the evaporator capacity decreases when refrigerant-oil solubility increases [33] due to the reduction in the enthalpy change across the evaporator.

Conclusions

In general, to obtain a high COP, combinations of high values of latent heat, liquid thermal conductivity and vapour density and low values of liquid viscosities and molecular weight are required. Trade-offs between capacity and COP should be considered for critical temperature and vapour specific heat.

It was noted that the evaluation of properties/parameters impact on the system performance also depends on whether the same equipments are used or not. For new system, a refrigerant with a higher volumetric refrigerating capacity (VRC) could use a smaller compressor for a given capacity. For drop-in or retrofit, the VRC of alternative fluids should be similar to that of the replaced fluid to obtain a similar capacity without changing the compressor.

From sensitivity point of view, the normal boiling point, the critical temperature, the liquid thermal conductivity and viscosity, the vapour density and the evaporative HTC appeared to have the significant impact on system performance.

In general, the presence of oil could decrease HTC and pressure drop, nevertheless it depends on the amount of oil and the solubility. The consequence of the change in HTC and pressure drop can affect the system performance.

Nomenclatures

c_p	isobaric specific heat [kJ·kg^{-1}·K^{-1}]
COP	coefficient of performance (= Q_{cool}/W_{com}) [---]
HX	heat exchanger
HTC	heat transfer coefficient [W·m^{-2}K^{-1}]
k	thermal conductivity [W·m^{-2}·K^{-1}]
MW	molecular weight [kg/kmol]
T	temperature [°C or K]
Δh_{refrig}	refrigerating effect [kJ·kg^{-1}]
ΔP	pressure drop [kPa]
VRC	volumetric refrigerating capacity or effect (= $\Delta h_{refrig}/v_{suc}$) [kJ/m^3]
ρ	density [kg·m^{-3}]
μ	dynamic (or absolute) viscosity [Pa· s]
v	specific volume [m^3·kg^{-1}]

Subscripts

c, cond	condenser or condensation or conduction
dew	dew (saturated vapour state)
e, evap	evaporator, evaporation
lat	latent
r	refrigerant

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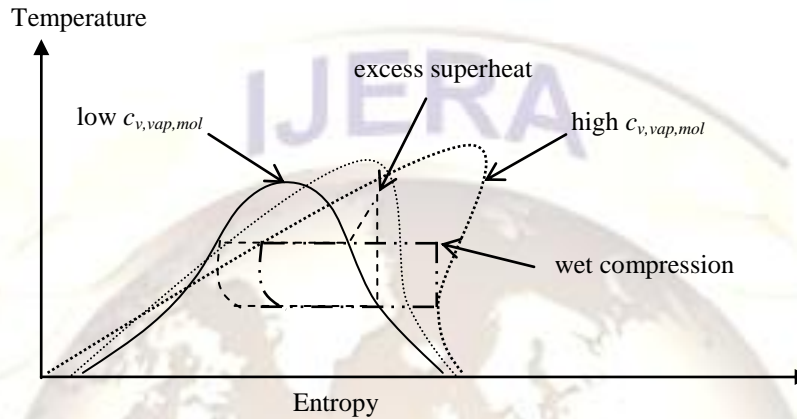
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Temperature

excess superheat

low $c_{v,vap,mol}$ high $c_{v,vap,mol}$

wet compression

Entropy

Figure 1 Effect of vapour heat capacity on the shape of the two-phase region or "vapour dome" on a temperature-entropy diagram (as adapted from Didion, 1999)

Table 1 Summary of desirable values of refrigerant properties for high *COP*

Refrigerant properties	Recommended values	
	High	Low
Latent heat	x	
Critical temperature	x	
Liquid thermal conductivity	x	
Vapour density	x	
Vapour specific heat	x	
Liquid viscosity		x
Molecular weight		x

Table 2 Variations in system performance caused by changes in thermo-physical properties and derived parameters in sensitivities analysis

Properties	Sensitivities	References
T_{crit}, T_b	T_{crit} and T_b ↑ 2 % => Q_{heat} ↓ 11-14 %, COP ↑ 0.3-1.8 % (vary depending on refrigerants); T_{crit} and T_b ↓ 2 % => Q_{heat} ↑ 11-14 %, COP ↓ 0.7-2.3 % (vary depending on refrigerants)	Hogberg and Vamling (1996)
c_p	$c_{p,vap}$ ↑ 5 % => Q_{heat} ↓ 1-4 %, COP ↓ 0.5-1.2 %, $c_{p,vap}$ ↓ 5 % => Q_{heat} ↑ 1-4 %, COP ↑ 0.5-1.2 %	Hogberg and Vamling (1996)
μ	μ_{liq} ↑ 15% => Q_{heat} ↓ 0.6-0.7 %, COP ↓ 0.4-0.5 % μ_{vap} ↑ 15% => Q_{heat} and COP ~ no significant change	Hogberg and Vamling (1996)
	μ_{liq} ↑ 50 % => Q_{cool} ↓ 4 %, W ↓ 1%	Domanski and Didion (1987)
k	k_{liq} ↑ 10 % => Q_{heat} ↑ 0.5-0.6 %, COP ↑ 0.5-0.6 % k_{vap} ↑ 15 % => Q_{heat} and COP ~ no significant change	Hogberg and Vamling (1996)
	k_{liq} ↑ 50 % => Q_{cool} ↑ 3 %, W_{com} ↑ 0.8 % k_{liq} ↓ 50 % => Q_{cool} ↓ 7 %, W_{com} ↓ 1.6 %	Domanski and Didion (1987)
ν	ν_{liq} ↑ 10 % => Q_{cool} ↓ 0.5 %, W_{com} ~ no change ν_{vap} ↑ 10 % => Q_{cool} ↑ 0.3 %, W_{com} ↓ 6 % ν_{vap} ↓ 10 % => Q_{cool} ↓ 0.5 %, W_{com} ↑ 7 %	Domanski and Didion (1987)
HTC_{evap}	HTC_{evap} ↑ 30 % => Q_{heat} ↑ 2.2 %, COP ↑ 1.8 % HTC_{evap} ↓ 30 % => Q_{heat} ↓ 4 %, COP ↓ 2.5 %	Hogberg and Vamling (1996)
	HTC_{evap} ↑ 50 % => Q_{cool} ↑ 2.5 %, W_{com} ↑ 1 % HTC_{evap} ↓ 50 % => Q_{cool} ↓ 6 %, W_{com} ↓ 2.5 %	Domanski and Didion (1987)
HTC_{cond}	HTC_{cond} ↑ 30 % or ↓ 30 % => Q_{heat} and COP ~ no significant change	Hogberg and Vamling (1996)
	HTC_{cond} ↑ 50 % => Q_{cool} ↑ 0.5 %, W_{com} ↓ 0.5 % HTC_{cond} ↓ 50 % => Q_{cool} ↓ 2 %, W_{com} ↑ 1.5 %	Domanski and Didion (1987)
$(\Delta P/\Delta z)_{evap}$	$(\Delta P/\Delta z)_{evap}$ ↑ 20 % => Q_{heat} ↓ 0.6 %, COP ↓ 0.2 % $(\Delta P/\Delta z)_{evap}$ ↓ 20 % => Q_{heat} ↑ 0.6 %, COP ↑ 0.1 %	Hogberg and Vamling (1996)
	$(\Delta P/\Delta z)_{evap}$ ↑ 100 % => Q_{cool} ↓ 2 %, W_{com} ↓ 0.8 % $(\Delta P/\Delta z)_{evap}$ ↓ 50 % => Q_{cool} ↑ 1 %, W_{com} ↑ 0.3 %	Domanski and Didion (1987)
$(\Delta P/\Delta z)_{cond}$	$(\Delta P/\Delta z)_{cond}$ change from −50 to 100 %, Q_{cool} and W_{com} ~ no significant change.	Domanski and Didion (1987)

Notes: 1) The numbers shown in the table are derived from the graphical results in the published literature. 2. Domanski and Didion (1987) studied only one refrigerant (R22), while Hogberg and Vamling (1996) studied R134a, R600a, R152a and R22/R142b.