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DECARBONISATION THROUGH INNOVATION

Extending the use of water-as-refrigerant based air-to-water heat pumps : solutions with a water-carbon dioxide cascade and water-ethanol mixtures

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Abstract

Recent regulatory changes concerning refrigerants are driving significant changes in the field of heat pumps (HP). The F-gas regulation requires a reduction in the use of hydrofluorocarbons (HFC). There is also a strong desire among certain European countries to tighten regulations on per- and polyfluoroalkyl substances (PFAS), also known as forever chemicals. Hydrofluoroolefin (HFO) refrigerants replace partially HFCs. However, since HFO decomposition products are part of PFAS, the risk of restrictions on their use in the future should not be overlooked. There are few alternatives to HFO. On the one hand, there are hydrocarbons, whose use is limited by their flammability. On the other, there are natural fluids. Water is one of them, and recent developments in high-speed electric motors have led to the emergence of compact high speed compressors with high volume flow rates.

However, a HP using water alone is limited to temperate climates with mild winters due to the relatively high triple point temperature of water.

In order to expand the range of uses for water as a refrigerant, the authors are working on two innovative solutions: a water-carbon dioxide cascade and water-ethanol mixtures. This article presents the modeling work on these two solutions. Regarding the cascade, the work focused on multi-variable optimization of the cycle. The results show very good HP performance for simultaneous heating and domestic hot water production (DHW). At 0°C ambient temperature the combined COP is 2.9°C for 47°C to 55°C heating regime and 55°C DHW production. In the case of water-ethanol mixtures, the work focused in particular on modeling fractionation in direct contact exchangers of the HP. The results show that using a mixture increases the COP and the specific capacity of the HP. The COP improvement is around 20%.

This work was carried out as part of the transnational SAFEHEAT project, which aims to produce proofs of concept of the mentioned solutions.

Keywords: natural refrigerant; thermodynamic cascade; refrigerant mixture.

1. Introduction

Heating buildings is responsible for around 10% of global CO_2 emissions [1]. In Europe, one third of natural gas imports are used to heat buildings [1].

The heat pump is a proven and sustainable technology for heat production.

It is particularly useful in decarbonizing the heating sector in countries with a low-carbon electricity mix, but also for countries with a mix more heavily dependent on fossil fuels. Indeed, the order of magnitude of the reduction in CO_2 emissions induced by replacing a gas boiler with a heat pump is 20% for countries like Poland, and around 80% for countries like France. So it's only natural that national plans to reduce greenhouse gas emissions set high ambitions for the deployment of heat pumps. In its Announced Commitment Scenarios (APS), the IEA estimates that heat pumps will account for half of global reduction in fossil fuel use for heating in buildings in 2030. In Europe, this requires an increase in installed capacity from 197GW in 2021 to 449GW in 2030 in the APS scenario. This represents the installation of around 30 million additional heat pumps compared to 2022.

In addition, it is now necessary to reduce Europe's dependence on fossil fuels, and in particular on imported natural gas, widely used to heat buildings. Replacing gas-fired boilers with heat pumps is therefore a fundamental lever in helping to achieve this reduction. Indeed, European Council is counting on the rapid deployment of heat pumps as part of its REPowerEU program, which aims to reduce its dependence on Russian gas [2]. The European APS scenario would result in natural gas savings of 20bcm which represents around 15% of current European consumption [1].

The regulatory environment in the heat pump sector is currently favourable to innovation. HFC refrigerants, introduced in response to the Montreal agreements on the ozone layer, are gradually being banned by the Kigali Amendment, which sets a target of reducing their production by 85% in 2036 compared to 2015. Indeed, it is estimated that HFCs account for 2.5% of global greenhouse gas emissions, due to fluid leakage and end-of-life management problems. Their ban has recently been reinforced in Europe, setting a reduction in their use to 95% of 2015 levels by 2030.

It is therefore essential to find alternatives to HFCs that are efficient, safe and quick to deploy. Alternatives to HFCs include HFO-type fluids, hydrocarbons and natural fluids.

HFO fluids are synthetic refrigerants that have been introduced in all refrigeration sectors since the 2010s. They are used, for example, in all new automotive air-conditioning systems (R1234yf in particular). Their global warming potential (GWP) is relatively low, thanks to their short atmospheric lifetime. In fact, once released, they rapidly degrade to give rise to TriFluoroacetic Acid (TFA), which is extremely persistent and a danger to the environment and human health. It is one of the synthetic chemicals of the per-and-polyfluoroalkyl (PFAS) group. Their level of danger is still under study, but they are strongly suspected of reducing fertility and increasing the risk of certain cancers, among others [3]. In this respect, five European countries are proposing a ban on PFASs under the REACH directive. This resulted in a first stage public consultation, which closed in September 2023, and will now be analyzed by the European Chemicals Agency (ECHA). The future of HFO synthetic fluids is therefore uncertain, both in terms of regulations and social acceptability.

Among natural refrigerants, **hydrocarbons** are an interesting alternative, but their use is restricted by European standards as they are classified as extremely flammable (class A3). For information, substances are classified as A3 if the concentration in air sufficient to generate combustion is less than 3.5% [4]. Hydronic monobloc systems can provide a solution to this problem, by placing the gas-containing unit outdoors for domestic applications. However, for larger installations such as collective housing, outdoor installation can be complex, and size restrictions lead to the use of series units containing a maximum of 5kg for example for propane (according to IEC 60335-2-40 standard), this represents a capacity of around 70kW, which represents a sub-optimal solution. From the point of view of industry professionals, the use of flammable refrigerants is controversial among some stakeholders [4]. The safety issues involved in handling class A3 fluids and the consequences in terms of the cost of additional safety measures, maintenance and specific training for installers are factors that could limit the speed of deployment of their use.

Other natural fluids must be investigated. Ammonia is an interesting fluid, however it is extremely dangerous because of its toxicity and flammability. It can be used in industrial environment with adequate safety standards; however, it is non-adapted to a residential use.

CO_2 and water are a promising alternatives and stand out for their non-toxicity and non-flammability.

CO_2 (R744) is widely used in commercial refrigeration today. The performance of CO_2 transcritical cycles to produce Domestic Hot Water (DHW) can be very high (around 4.3 for a 9°C/60°C regime at 0°C evaporation) because in this case, the cycle can take advantage of the sub-cooling provided by the fresh water needed for DHW even if CO_2 operates in supercritical regime. This is not possible for heating, as in this case there is no possibility of sub-cooling by any free source and the performance drops. For CO_2 heating to be more effective, it should operate in subcritical mode, i.e. at condensation temperatures below 30°C; under these conditions, even at sub-zero ambient temperatures, CO_2 offers good very efficiencies. For this reason, some commercial refrigeration cascades use CO_2 for cooling capacity production topped by an ammonia cycle [5].

Water as a refrigerant (R718) is an emerging solution. In addition to its toxicological and climatic neutrality, water has exceptional thermodynamic and thermal characteristics. However, for a residential heat pump cycle, the most common source being ambient air, the evaporator operates at relatively low pressures. Table 1 gives water vapour pressures and densities as a function of temperature.

The very low density values explain why Leviathan Dynamics and some other companies have decided to develop very high-speed centrifugal compressors. This was made possible by the recent technical development of high-speed electric motors which has been developed through the development of high-speed machining.

Table 1: water vapor properties

T (°C)	5	10	30	50	70	80	90	100
P (bar)	**0,009**	0,012	0,042	0,124	0,312	0,474	0,702	1,014
ρ (kg/m^3)	**0,007**	0,009	0,03	0,083	0,198	0,294	0,424	0,598

The challenge is therefore to produce compressors of modest size, for reasons of integration and price. Leviathan Dynamics has developed a centrifugal compressor with very high rotational speeds, 90 kRPM, enabling previously impossible compactness to be achieved. A Leviathan compressor model is shown in **Figure 1**.

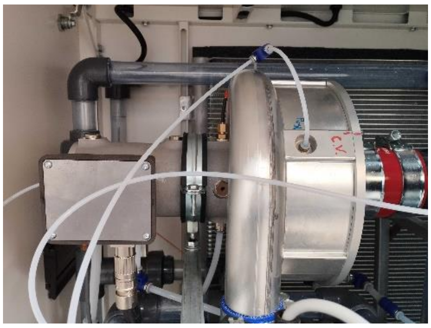

Figure 1 : Leviathan compressor

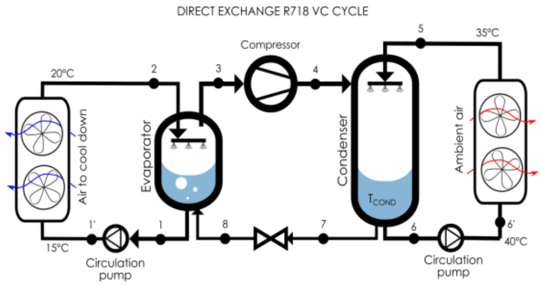

Figure 2 : R718 direct heat exchangers cycle; air-cooling case

Leviathan has also developed and improved direct-contact condensers and evaporators in collaboration with the CEEP from Mines Paris PSL, enabling very low approach temperatures to be achieved between the working fluid and the heat transfer fluid, which in both cases is water [6]. Leviathan Dynamics' systems have COPs of the order of 8 for air-conditioning applications, opening up fabulous development perspectives. The components architecture currently used by Leviathan is shown in **Figure 2**. The evaporator and the condenser are direct contact heat exchangers that perform very high heat transfer coefficients. Both are connected to hot and cold sources respectively using classical heat exchanger and a liquid loop.

Unfortunately, water is limited by its solidification point at 0°C. Also, at low evaporation temperatures, the compression ratios required for heating would be enormous, which is technically complex to achieve.

In the present article two ways of extending the operational range of water-as-refrigerant based systems are presented:

- A R718/R744 thermodynamic cascade. The R744 operates always in sub-critical state in the bottom cycle; it can efficiently operate at negative evaporating temperatures. The R718 operated as topping cycle. This combination takes advantage of operating both fluids in conditions that maximize their respective thermodynamic advantages and avoiding or limiting their drawbacks.
- A water/ethanol mixture. The goal is to lower the freezing temperature of the refrigerant without penalizing its evaporation pressure. Besides the mixture has higher vapour density and lower specific heats ratio which helps in achieving more compact and efficient systems.

2. Water /carbon dioxide cascade

2.1. Cascade description & operating conditions

The R718/R744 cascade benefits from the thermodynamic advantages of water at typical heating temperatures and those of CO_2 at low temperatures.

Figure 3 shows the cascade layout. The CO_2 cycle supporting the water cycle is subcritical. This is a key aspect of the cascade since it allows operating the CO_2 cycle at very high performances. The two cycles are coupled by an evapo-condenser. This type of system would also enable DHW to be produced simultaneously with heating water, as the energy from CO_2 de-superheating at high temperatures can easily be recovered.

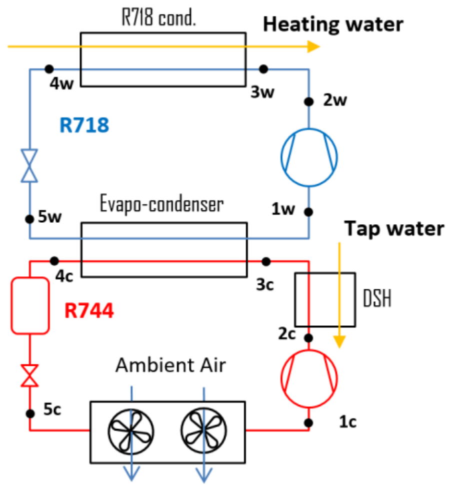

Figure 3 : R718/R744 cascade components arrangement

Three operating modes are analysed on this cascade. Table 2 presents them. For each mode a series of variants were analysed.

Table 2: R718/R744 system operating modes

Full Heating	**Mixed Mode**	**Full Domestic Hot Water Production**
Heating : 30°C / 35°C	DHW: 10°C/ 55°C Heating : 30°C /35°C	DHW: 10°C/ 55°C • Cascade with parallel water HXs • Supercritical CO_2
Heating : 47 °C / 55°C	DHW: 10°C/ 55°C Heating : 47°C / 55°C	

In the full heating mode, both R718 condenser and R744 de-superheater (DSH) provide heating. This mode represents a period of the day where DHW need has been satisfied. In mixed mode the R718 condenser provides heating and the de-superheater provides DHW. The last mode is full DHW production and represents a summer operating condition for example. This mode can be performed with several architectures. The two best architectures are presented here: the cascade with in-parallel DHW water circuit and R744-only supercritical cycle. In the supercritical CO_2 situation, only the CO_2 system is used and the power of the Evapo-condenser is zero.

Regarding the temperatures for heating, classical levels of low temperature emitters and medium temperature emitters were selected (30°C→35°C and 47°C→55°C). For DHW, 10°C inlet temperature and 55°C outlet temperature were selected.

Regarding the ambient temperatures, -5°C, 0°C and 5°C were selected. In future works it is planned to use the meteorological data from Helsinki, Copenhagen and Paris.

The temperature difference between air inlet and outlet is 5K.

2.2. Modeling assumptions and optimization constraints & variables

A certain number of assumptions were made to define the thermodynamic states of the R718/R744 system. The cascade is modeled is steady state assuming no pressure losses. Besides, the cycle implemented by Leviathan Dynamics being based on direct contact heat exchangers, a liquid circuit connects the evaporator and condenser respectively to cold and hot sources, which results in two stages of heat transfer devices for each loop; this is included in the model by assuming an adapted level of approach temperature difference (also named pinch temperature difference). Some other basic parameter necessary to define the cycle are given in Table 3.

Table 3: cascade fixed parameters

Parameter	Retained values
Sub-cooling degree at the condensers outlet (SC)	0K for both fluids (state : liquid)
Superheating degree at evaporators outlet (SH)	10K for R744 and 4K for R718
Evapo-condenser pinch ($\mathbf{dT_{pinch-ev/cond}}$)	3K
R744 evaporator pinch ($\mathbf{dT_{pinch\_evR744}}$)	5K

The third parameter gives the temperature difference between the R744 condensation temperature and R718 evaporation temperature. The value was chosen to represent a falling film type evapo-condenser.

The last parameter of the table is used to define the evaporation temperature and thus the evaporation pressure using expression (1).

$$T_{ev\_R744} = T_{air\_in} - dT_{pinch\_evR744} - SH_{R744} \tag{1}$$

The isentropic efficiency of the R744 compressor is defined according an expression from relationship of Nebot-Andrés et. Al, 2017 [7]. It is relying on R744 compression ratio ($\tau = P_{cond_{CO_2}} / P_{evap_{CO_2}}$).

$$\eta_{comp_{CO2}} = 0.736 - 0.052 \ \frac{P_{cond_{CO2}}}{P_{evap_{CO2}}} \tag{2}$$

The R744 outlet state is then:

$$h_{2c} = \frac{h_{2cs} - h_{1c}}{\eta_{comp_{CO2}}} + h_{1c} \tag{3}$$

The isentropic efficiency of the R718 compressor is 0.55 and comes from Leviathan Dynamics experimental values.

Regarding the R718 condenser, the approach temperature is treated as a constraint. The pinch point is assumed to be located on the dew point (eq. (2)). The respective heating water temperature ($T_{heat\_water\_pinch}$) is computed by a power balance on the R718 condenser.

$$T_{dew\_cond\_R718} - T_{heat\_water\_pinch} = dT_{pinch\_condR718} \tag{4}$$

For a given heating regime, the R718 condensing pressure being dependent on the pinch temperature difference, an iterative approach is used to find it.

Regarding the temperature regime in the R744 de-superheater, it depends on the mode. The hot side and cold side temperature differences are checked in order to satisfy a 3K pinch temperature difference. For all cases the following constraint equations are verified:

$$T_{DSH\_out\_hot} - T_{DSH\_in\_cold} \geq 3K \tag{5}$$

$$T_{DSH\_in\_hot} - T_{DSH\_out\_cold} \geq 3K \tag{6}$$

To end the cycle definition two variables need to be defined:

- the R718 evaporating temperature ($T_{ev\ R718}$)
- the DSH R744 outlet temperature ($T_{DSH\ out\_hot}$)

They are here optimized in order to maximize the Coefficient Of Performance (COP).

The evaporation temperature will strongly affect the respective performances of both R718 and R744 cycles and has an optimal value between 5°C and 25°C depending on the case. The second variable gives the relative power that is transferred trough the R718 condenser and the DSH and affect the COP since it defined the relative capacities of the R718 and R744 compressors. For a reason of facilitating the mathematical formulation and algorithmic handling of this parameter ($T_{DSH\_out\_hot}$), it is defined as function of the R744 condensation temperature as follows:

$$T_{DSH\_out\_hot} = T_{cond\ R744} + dT_{marg} \tag{7}$$

In the solver the dT_{marg} is the one treated as an unknown. It can take any value between 0K and 20K.
An example of the effects of the constraints on the variables is the case of low optimal values of $T_{ev\ R718}$. In some cases, the optimal value of $T_{ev\ R718}$ can take values below 10°C which forces the solver to attribute non optimal values to dT_{marg} in order to guaranty the constraint of eq. 6.

In the case of supercritical R744 cycle for DHW production, the variable to be optimized is the gas cooling pressure. The gas cooler temperature profile is used to compute the pinch constraint that can appear at any location including the outlet.

The COP is defined as follows:

$$COP = \frac{\dot{Q}_{DSH\_R744}+\dot{Q}_{cond\_R718}}{\dot{W}_{comp_{R744}}+\dot{W}_{comp_{R718}}} \tag{8}$$

The modeling was done on python using the “Coolprop” library to obtain the thermodynamic states of the cycle. The optimization was done in python using various approaches among them the “minimize” function on the “scipy” library. These approaches are multivariable one-objective methods. The inverse of the COP is used as objective to be minimized.

2.3. Optimization results

Table 4 presents the optimization results. The table gives the COP, $T_{ev\ R718}$, dT_{marg} and τ_{R718} (the R718 compression ratio). It gives also for the “Full DHW” mode the results for the supercritical R744 case.

Table 4: R718/R744 system optimization results

		Full Heating		**Mixed**		**Full DHW**	
T_{amb}	Case	30°C/35°C	47°C/50°C	30°C/35°C	47°C/50°C	Cascade	R744 supercritical
-5°C	COP	3.13	2.46	3.26	2.70	2.86	2.94
	$T_{ev\ R718}$	4°C	10°C	6°C	12°C	7°C	$T_{GC\ out}$= 15°C
	dT_{marg}	26°C	37°C	6°C	0°C	5°C	P_{GC}= 76 bars
	τ_{R718}	7.8	13.5	6.8	11.9	11.167	N/A
0°C	COP	3.41	2.61	3.55	2.9	3.08	3.30
	$T_{ev\ R718}$	7°C	12°C	11°C	13°C	11°C	$T_{GC\ out}$ = 14°C
	dT_{marg}	23°C	35°C	2°C	0°C	2°C	P_{GC}= 78 bars
	τ_{R718}	6.3	11.9	4.9	11.2	8.9	N/A
5°C	COP	3.74	2.77	3.88	3.13	3.32	3.71
	$T_{ev\ R718}$	9°C	14°C	13°C	15°C	15°C	$T_{GC\ out}$= 15°C
	dT_{marg}	21°C	33°C	3°C	0°C	0°C	P_{GC}= 79 bars
	τ_{R718}	5.5	10.5	4.3	9.8	7.1	N/A

Regarding the COP, the “mixed” mode has highest values. This is related to the beneficial effect of the DHW water stream to be heated-up that allows small margin temperature. This suggest that “mixed” mode needs no be prioritized when operating such a cascade. Regarding the “full DHW” mode, the supercritical R744 system has better COP. This is related to a better matching between DHW water stream and CO_2 temperature profiles.

Regarding the comparison of this solution to a ‘competitor’ solution like R290 based HPs, a full analysis was not made here, but as a first result, at 0°C of ambient temperature, the COP is 6% lower for the cascade at 30°C→35°C regime and 5% higher at 47°C→55°C regime. This was computed at equivalent pinch temperature difference conditions with dedicated R290 cycles for each service. This shows that the cascade has a great potential.

The R718 optimal evaporation temperature is relatively low in general, which illustrates the good efficiency of the R718 cycle. The higher the heating temperature, the higher the R718 evaporation temperature especially for "full heating mode".

The optimal margin temperature (dT_{marg}) is highly affected by the presence of DHW. When no DHW is required, the DSH heat exchanger being used to provide also heating, in order to keep $T_{ev\ R718}$ sufficiently low, the optimal solution respecting the pinch constraints leads to an increase of dT_{marg}.

The optimal compression ratios are relatively high and some of them are hardly attainable with centrifugal compressor. For this reason, in future, the technological constraint of attainable compression rate will be added in the process.

Figure 4 shows the temperature-entropy diagram of a "mixed" mode and 30°C→35°C heating regime. It shows as well the temperature regimes of heating water and DHW for -5°C ambient air temperature.

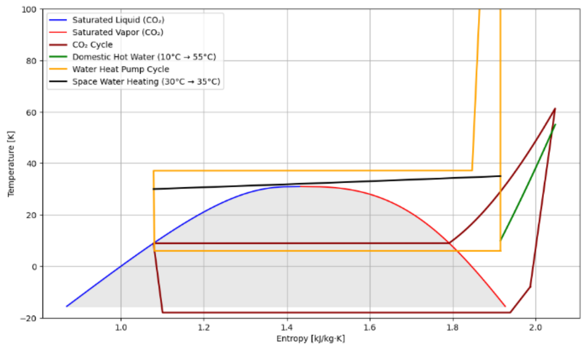

Figure 4: TS diagram of "mixed" mode, T_{amb} -5°C

2.4. Partial conclusion

The results for the R718/R744 cascade show that mixed mode, combining simultaneous heating and DHW production, presents the best COP, thanks to better matching of temperature levels in the CO_2 desuperheater. COP typically vary between 3 and 3.9 depending on the ambient conditions and heating regime. The supercritical R744 variant proves to be more efficient for DHW production alone, due to temperature profiles that are better suited to heating domestic water. Optimization highlights relatively low R718 evaporation temperatures, indicating good water cycle efficiency. On the other hand, R718 compression ratios remain high and sometimes exceed realistic limits for centrifugal compressors, which is a notable technological constraint. Finally, a preliminary comparison shows that the cascade is competitive with R290, with slightly lower performance at low temperatures but higher performance at higher temperatures.

3. Water/ethanol mixture

A solution to reduce the minimal operating temperature of R718 based system consists in adding ethanol. Ethanol can help to reduce compression ratios (by around 30% at a concentration of 10%) as well as the fluid's adiabatic index (by around 15%), thereby increasing the efficiency of the cycle.

3.1. Two types of cycles

Two types of cycles are presented here:

- Ideal. In the ideal cycle, the circulating composition is equal to the average composition. Figure 5 represents this case were no fractionation or any partial retention can occur. This requires using for example plate heat exchanger, which is not easy with R718 at temperatures below 80°C.
- Fractionated. In the fractionated cycle, a separation of the mixture happens because of the specific architecture of the Leviathan Dynamics direct contact heat exchangers. In such heat exchangers the vapor and liquid would have different ethanol contents leading the compressor to compress a mixture with higher content of the most volatile compound i.e. ethanol.

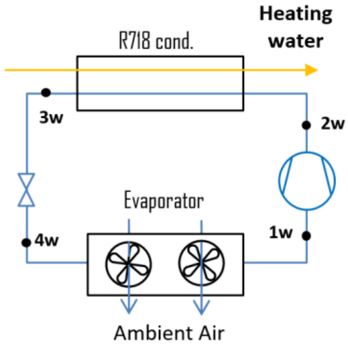

Figure 5: ideal cycle

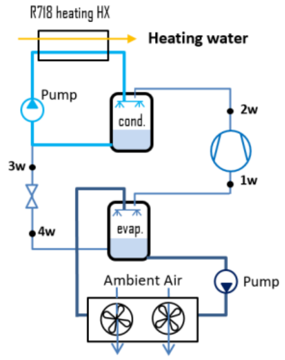

Figure 6: real cycle

Both cycles were evaluated in order to quantify the eventual relative advantages of both cycles.

3.2. System modeling

3.2.1. Mixture properties

The mixture model has to estimate correctly the thermodynamic states of the refrigerant. Eventually it should estimate correctly the triple point temperature of the mixture. This point is not obvious to guarantee and when using “Coolprop” library, this parameter is not accessible. Thus the freezing temperature function of the ethanol concentration was implemented in the model based on multi-linear model using the data from [8] presented in Figure 7.

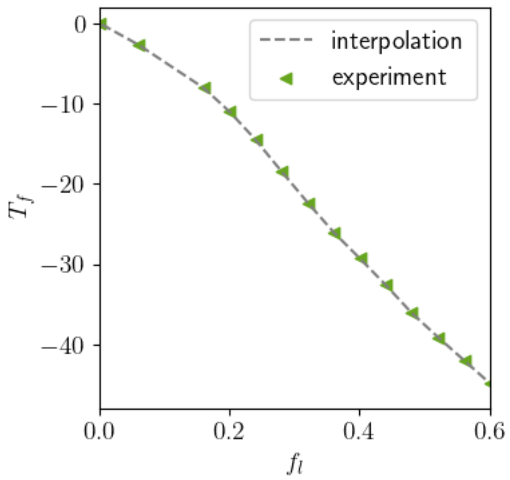

Figure 7: water ethanol mixture freezing temperature [8]

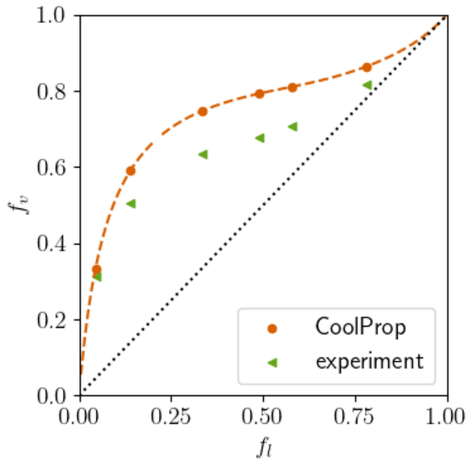

Figure 8: Ethanol mole fraction in vapour phase (f_v) as function of ethanol mole fraction in the liquid phase (f_l) at 25°C [9]

Figure 8 presents an example from the Coolprop library compared to vapor-liquid equilibrium experimental data reported in [9] and obtained by Dobson; it shows the vapor phase ethanol molar concentration as a function of liquid phase ethanol concentration at 25°C. This shows that a discrepancy of 15% exist in molar fraction at liquid ethanol fractions close to 50%. This highlights the necessity of improving the mixture models and should do in the future. For the present work, the Coolprop mixture data will be used as a preliminary approach.

3.2.2. Cycles modeling and assumptions

The thermodynamic states definition for the ideal cycle is similar to the cascade cycle in terms of sub-cooling, superheating and pinches definition. Thus, the thermodynamic states were obtained using the average ethanol concentration to be analyzed. Only heating mode for both 30°C→35°C and 47°C→55°C regimes was considered here.

However, regarding the real cycle the fraction need to be included in the analysis to attribute the correct composition to each point of the cycle. The main conservation assumptions are that compressor inlet concentration is equal to evaporator vapor concentration ($f_{ev\ g}$) and that expansion valve concentration is equal to the concentration in the liquid of the condenser ($f_{co\ l}$) as expressed by the following equations:

$$f_{1W} = f_{ev\ g} \tag{9}$$

$$f_{3W} = f_{co\ l} \tag{10}$$

Besides, at equilibrium:

$$f_{1W} = f_{3W} \tag{11}$$

Then for a given average initial composition and given the volumes of evaporator and condenser of the real device, the system is solved using python.

3.3. Modeling results

The results of the real system are provided here first to illustrate the differences between a pure water system and a mixture based system.

Figure 9 shows the COP and the volume flow rate as function of ambient temperature for 0% and 50% ethanol mass fractions. The volume flow rate is computed for a target capacity of 10kW heating capacity ($D_m = \dot{Q}/(\rho_{1W}\Delta h_{ev})$).

The COP is improved when considering a mixture. For the lowest temperature water regime the improvement is around 20%. Besides, the minimal temperature that can be operated in the case of pure water case is limited to a temperature close to 10°C when considering the pinch and the ambient air temperature profile. In the case of the mixture, the minimal temperature can be much lower; the minimal value is discussed later.

Regarding the required volume flow rate, it is lower for the mixture than for pure water. This results from an increasing vapor density despite a reduced evaporation enthalpy. The result is that for a given compressor the capacity will increase.

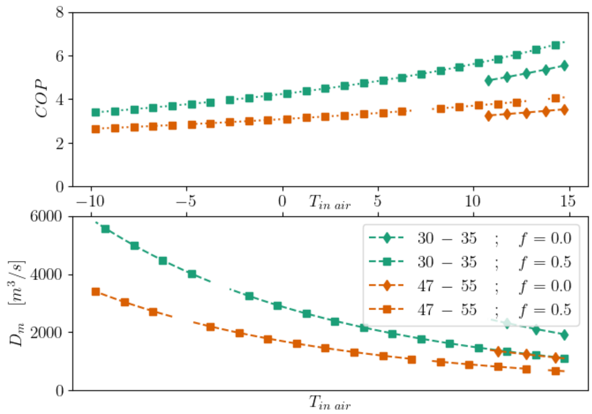

Figure 9: COP and volume flow rate as function of ambient temperature for 0% and 50% ethanol mass fractions

Figure 10 gives the COP as function of the minimal allowed temperature (T_{min}) for the ideal and real cycles for the two heating regimes. The value of T_{min} is obtained by the ethanol circulating mass fraction in the stream in contact with lowest air temperature which is the most critical point of the cycle. To obtain the corresponding ambient temperature the reader may add 12K to T_{min}.

The figure gives also the ethanol concentration for each section of the curves. The maximum tested concentration was 55%. The black symbols indicate the concentration for corresponding curves.

The COP obtained is better with the real cycle. However, the minimal temperature that can be attained for a given average ethanol concentration is lower with the ideal cycle. This comes from lower ethanol concentrations that circulates in the liquid side of the evaporator in the real cycle since for a given average ethanol concentration, when including fractionation, the liquid contains lower ethanol concentrations.

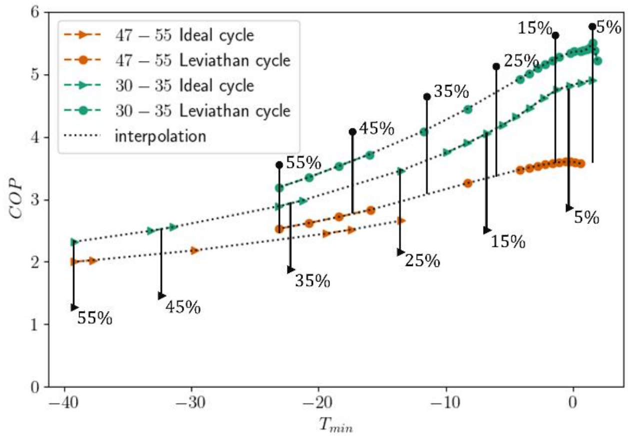

Figure 10: COP function of the minimal allowed temperature

3.4. Partial conclusion

The results for water/ethanol mixtures indicate a net gain in thermodynamic performance compared to pure water, with a higher COP and a reduced volume flow rate, which is beneficial for compressor compactness. The addition of ethanol lowers the minimum operating temperature and increases vapor density, extending the range of use to colder climates. The actual cycle, including fractionation in direct heat exchangers, performs better than the ideal cycle because the ethanol-enriched vapor improves compression conditions. However, this fractionation limits the minimum achievable temperature, as the liquid phase becomes too low in ethanol to remain liquid at low temperatures. These results highlight the real interest of water/ethanol mixtures in extending the feasibility of R718 cycles in cold conditions, subject to a more accurate mixture model.

4. Conclusion

The R718/R744 cascade and water-ethanol mixture show significant advantages in performance and operating range compared to pure water. The cascade optimization showed the interest of the “mixed” mode that provides simultaneously heating and DHW services. A next step in the assessment of this solution is including technological constraints related to the feasible compression ratios depending on the technology selected. Besides a test bench is under development to test the cascade in mixed mode; this will provide very useful data to validate the results obtained theoretically. A first comparison to a R290 system shows that the solution is competitive while offering a solution using non-flammable refrigerants.

Regarding the ethanol-water mixture, the analysis shows the interest of adding ethanol since this leads to higher COP and specific capacities. However, the equations of state used give some discrepancies with experimental equilibrium date; a next step will be to model the cycle with better equations of state models (for example using ASPEN).

This work contributes to the development of safer heat pump systems that are compatible with future regulations.

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