# Impact of Oil Solubility and Refrigerant Flashing on the Performance of Transcritical $CO_2$ Vapor Compression Systems with Oil Flooding and Regeneration

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## ABSTRACT

Flooding the compressor of a vapor compression system with oil can allow for a more isothermal compression process. This can lead to significant improvements in performance, particularly when combined with a regenerative heat exchanger. For $CO_2$ cycles with supercritical heat rejection, the superheat horn and throttle are major sources of irreversibility. The combination of flooding and regeneration attacks both of these losses with a relatively small impact on system costs. The improvement in system COP can be over 20% for sink and source temperatures of 30°C and -30°C, respectively, with an optimized oil mass fraction and oils that have low refrigerant solubility. However, when some amount of refrigerant is solved in the oil, the cooling capacity is decreased. At a refrigerant solubility of 20% by mass in the oil for fixed oil mass fraction, there is no improvement in system COP compared to the baseline system.

## 1. INTRODUCTION AND BACKGROUND

Hugenroth (2006) and Hugenroth et al. (2006, 2007) investigated flooding of scroll compressors and expanders as a means of realizing a practical Ericsson cycle. The Ericsson cycle is a theoretical thermodynamic cycle that can achieve the same efficiency as the Carnot cycle. The Ericsson cycle consists of isothermal compression and expansion processes in addition to reversible regeneration. A number of means have been proposed to achieve isothermal compression, including cooling the compressor during the compression process (Kim et al. 2008).

Quasi-isothermal compression can be achieved through the addition of a large amount of high specific heat fluid to the refrigerant flow to absorb the heat of compression of the refrigerant. In practice, lubricity considerations would almost certainly require that the fluid be liquid oil, but other fluids are possible. Hugenroth (2006) and Hugenroth et al. (Hugenroth 2006, Hugenroth et al. 2007) have extensively studied liquid flooding used to approximate the Ericsson cycle, and also showed from simplified compressor modeling that the ideal flooding liquid was water due to its low specific volume and high specific heat.

In the last few decades, a number of screw compressor researchers have investigated flooding with oil in order to improve the sealing of the compressor (Blaise & Dutto 1988, Wu & Jin 1988). Recently a number of authors have applied oil flooding to refrigerant scroll compressors (Hiwata et al. 2002, Sakuda et al. 2001), for which some positive results have been found. Mechanistic modeling of the liquid-flooded compression and expansion processes in scroll compressors and scroll expanders has been carried out (Bell et al. 2008a, Bell et al. 2008b, Lemort et al. 2008), from which a thorough understanding of the impact of oil has been obtained.

A number of researchers have also considered the impact of oil flooding on the cycle performance of vapor compression systems. Hugenroth (2006b) considered the impact on a R410A air conditioning unit, though the analysis did not include the addition of a regenerator. Bell (2009) considered the impact of oil flooding on a wide range of working fluids, for which the most promising results were found with $CO_2$ and R404a. These promising results motivated a further study by Bell (2010) which investigated the impact on $CO_2$ cooling cycles in greater detail. Ignatiev et al. (2008) are the patent holder for a patent which covers a wide range of these oil flooding technologies.

## 2. FLOODED COMPRESSION

The technology of flooded vapor compression involves modifications to a standard vapor-compression cooling cycle that includes the addition of a regenerator and an oil-processing circuit, as seen in Figure 1. In this configuration, oil and refrigerant vapor are mixed adiabatically prior to injection into the compressor, and then are compressed together in the compressor from state point *1* to state point *2*. After exiting the compressor, the two-phase mixture of oil and refrigerant vapor enters the high-pressure separator at state point *2*, and an oil stream with some amount of refrigerant vapor solved and a pure refrigerant vapor stream exit the separator at state points *9* and *3,* respectively. The oil and refrigerant are cooled in parallel circuits in the condenser/oil cooler to near the heat sink temperature. If the condenser pressure is supercritical, as is the case here, the heat rejection will happen as quasi-single-phase heat transfer from the supercritical fluid. The oil exiting the oil cooler at state point *10* is expanded back to state point *11*, at which point it is mixed back into the refrigerant vapor exiting the regenerator. After being condensed in the condenser, the liquid refrigerant passes through a regenerator where it exchanges heat with the refrigerant exiting the evaporator to sub-cool the refrigerant exiting the condenser. The sub-cooled refrigerant is throttled through the expansion device and then evaporated in the evaporator.

As a result of flooded compression, the thermodynamic property plots for the refrigerant (with the baseline cycle refrigerant state points overlaid with dashed lines for comparison) exhibit some unique features, as seen in Figure 2. For one, the specific enthalpy of the gas decreases as the gas-oil mixture is compressed from low to high pressure (though the total mixture specific enthalpy increases due to the work of compression of the oil) due to the cooling of the refrigerant by the oil. Also, with the presence of regeneration, the gas liquid mixture will always enter the compressor at nearly the same temperature. This is due to the fact that the oil is cooled in the oil cooler to near the heat sink temperature. By design, the capacitance flow rate of the oil is significantly higher than that of the gas, and the gas will take on the oil temperature during the mixing process.

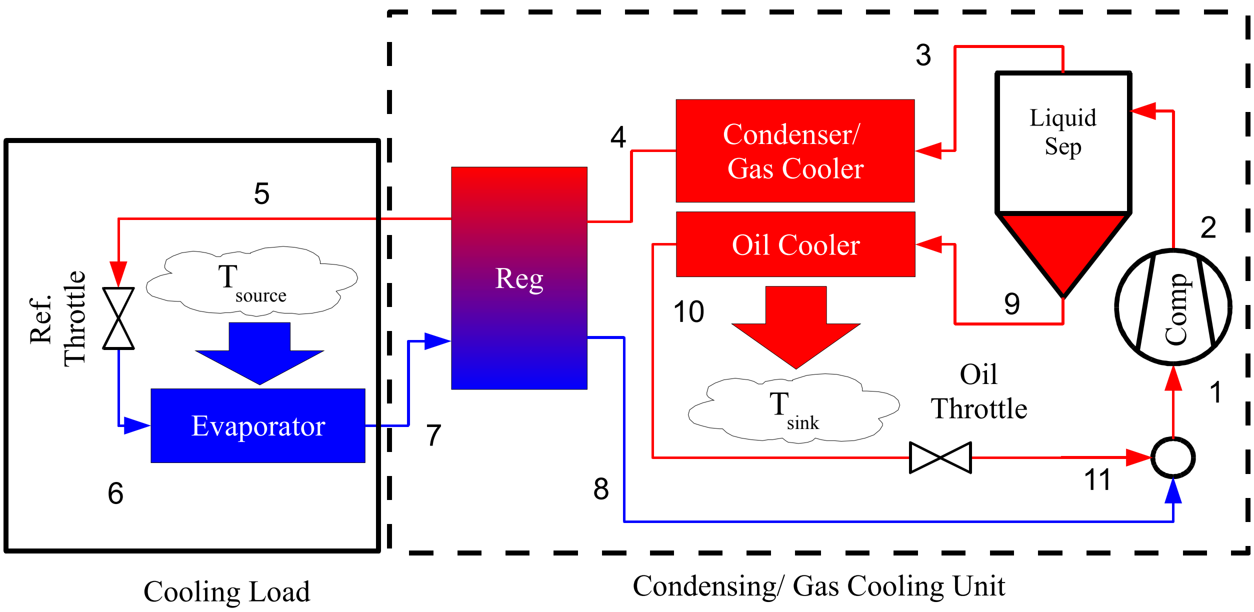



Figure 1 Flooded compression system schematic

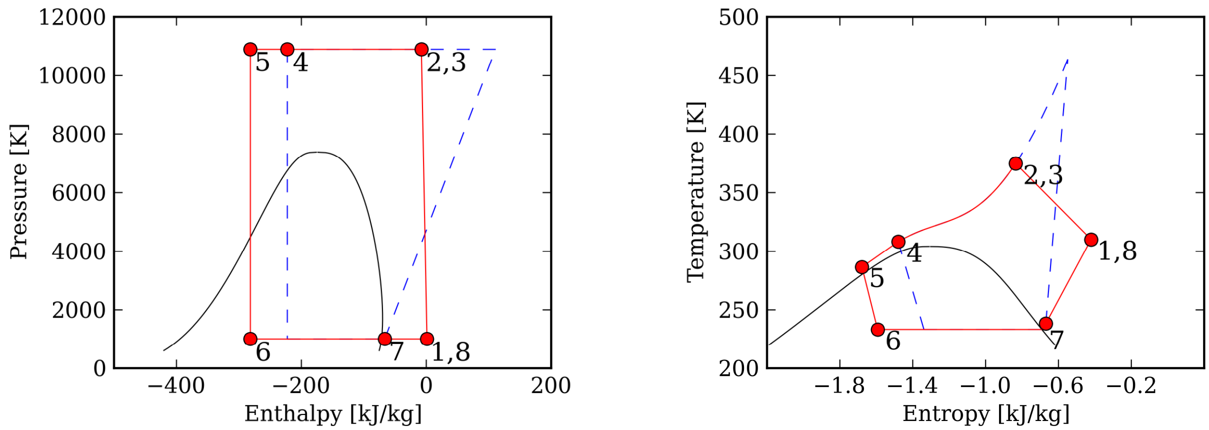



Figure 2 Flooded compression pressure-enthalpy and temperature-entropy plots
(Dashed lines: baseline cycle)

## 3. CYCLE MODELING

The analysis of the flooded vapor compression system with oil solubility follows that of Bell et al. (2009, 2010) for systems without oil solubility. The relevant equations are found in Table 1. As a result of the oil solubility, the primary difference is that not all the refrigerant which passes into the oil separator goes through the evaporator. In the separator, chemical equilibrium determines the solubility of refrigerant in the oil. The solved refrigerant then passes through the oil cooler where it is condensed out, and then expands through the oil throttling valve. From a cycle performance standpoint, it is best if the refrigerant has zero solubility in the oil, but this is not possible to achieve in practice. The parametric study presented here has been carried out in order to quantify the deleterious effect of refrigerant solubility in oil. In the separator, the total mass flow rate of gas and oil balance, and the amount of refrigerant leaving the separator solved in the oil is taken to be a fraction of the oil mass flow rate, which is the equivalent of the equilibrium solubility mass fraction. Figure 3 shows the mass balance over the oil separator. The oil mass flow fraction is defined by

$$x_L = \frac{\dot{m}_L}{\dot{m}_L + y_{solve}\dot{m}_L + \dot{m}_{g,v}} \tag{1}$$

which is an imposed variable in the model and allows for calculation of the refrigerant flow rate.

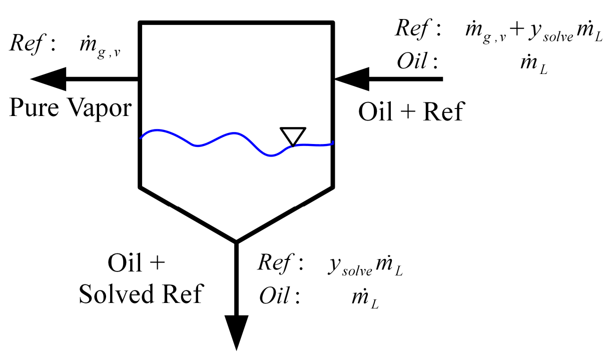



Figure 3 Schematic of Oil Separator

The following assumptions were used to simplify the cycle analysis:

- Refrigerant vapor and flooding liquid are in thermal and mechanical equilibrium (same temperature and pressure)
- The effect of solubility of refrigerant vapor in the flooding liquid is only considered in the oil separator and oil throttle
- Fixed superheat, subcooling, and pinch temperature between outlet of heat exchanger and air temperature
- Only transcritical operation considered at a gas cooler pressure that optimizes the efficiency of the flooded, no-solubility cycle
- Same fixed volumetric displacement rate of compressor for baseline and flooded systems with a volumetric efficiency of 100%
- Other fixed parameters as shown in Table 2
- Properties of polyalkyl glycol (PAG) oil used with constant specific heat

The equations in Table 1 were implemented in the Python programming language along with a solver routine to efficiently solve the system of equations. The Span and Wagner (1996) equation of state for $CO_2$ is used within the model to provide the thermodynamic properties.

The mixture properties are calculated based on an ideal mixture formulation, which yields:

$$\begin{aligned} h_m &= x_L h_L + \left(1 - x_L\right) h_G \\ s_m &= x_L s_L + \left(1 - x_L\right) s_G \end{aligned} \tag{2}$$

This formulation eliminates the complexities of phase equilibrium in the supercritical state and allows for a reasonable approximation to the actual system performance.

Table 1 Equations used to model Flooded Compression system

| Compressor | Condenser | Oil Separator | Oil Cooler / Oil Throttle |
|---|---|---|---|
| $h_{1m} = h\left(T_1, p_1, x_{L1}\right)$<br>$s_{1m} = s\left(T_1, p_1, x_{L1}\right)$<br>$h_{2sm} = h\left(p_2, x_{L2}, s_1\right)$<br>$\eta_{comp} = \dfrac{h_{2sm} - h_{1m}}{h_{2m} - h_{1m}}$<br>$\dot{W}_{comp} = \dot{m}_m\left(h_{2m} - h_{1m}\right)$<br>$\dot{\mathbf{E}}_{comp} = T_0 \dot{m}_m\left(s_{2m} - s_{1m}\right)$ | $T_4 = T_{\text{sink}} + \Delta T_{\text{pinch}}$<br>$h_4 = \begin{cases} h_{sat}\left(T_4, x=0\right) & p_4 < p_{crit} \\ h\left(p_4, T_4\right) & p_4 > p_{crit} \end{cases}$<br>$\dot{Q}_{cond} = \dot{m}_G\left(h_4 - h_3\right)$<br>$\dot{\mathbf{E}}_{cond} = T_0\left[\dot{m}_G\left(s_4 - s_3\right) - \dfrac{\dot{Q}_{cond}}{T_{sink}}\right]$ | $T_3 = T_9 = T_2$<br>$p_3 = p_9 = p_2$<br>$x_{L3} = 0$<br>$x_{L9} = \dfrac{1}{1 + y_{solve}}$<br>$\dot{\mathbf{E}}_{sep} = 0$ | $T_{10} = T_{sink} + \Delta T_{pinch}$<br>$\dot{Q}_{oil-cooler} = \dot{m}_L\left(h_9 - h_{10}\right)$<br>$h_{10} = h_{11}$<br>$\dot{\mathbf{E}}_{throttle} = T_0 \dot{m}_L\left(s_{11} - s_{10}\right)$ |

| Evaporator / Expansion Valve | Regenerator | Mixer |
|---|---|---|
| $T_{evap} = T_{source} - \Delta T_{pinch} - \Delta T_{sh}$<br>$h_7 = h_m\left(T_{evap} + \Delta T_{sh}, p_{evap}\right)$<br>$\dot{Q}_{evap} = \dot{m}_G\left(h_7 - h_6\right)$<br>$\dot{\mathbf{E}}_{evap} = T_0\left[\dot{m}_{g,v}\left(s_7 - s_6\right) - \dfrac{\dot{Q}_{evap}}{T_{source}}\right]$ | $\Delta h_{\text{reg,max}} = \min\begin{cases} h_4 - h\left(T_7, p_4\right) \\ h\left(T_4, p_7\right) - h_7 \end{cases}$<br>$\varepsilon = \dfrac{\Delta h_{\text{reg}}}{\Delta h_{\text{reg,max}}}$<br>$h_5 = h_4 - \Delta h_{\text{Reg}}$<br>$h_7 = h_6 + \Delta h_{\text{Reg}}$<br>$\dot{\mathbf{E}}_{reg} = T_0 \dot{m}_{g,v}\left[\left(s_8 + s_5\right) - \left(s_4 + s_7\right)\right]$ | $\left(\dot{m}_L + \dot{m}_{g,s}\right) h_{11} + \dot{m}_G h_8 - \left(\dot{m}_L + \dot{m}_G\right) h_{1m} = 0$<br>$\dot{\mathbf{E}}_{mixer} = T_0\left[\begin{array}{c} \left(\dot{m}_L + \dot{m}_G\right) s_{1m} - \dot{m}_G s_8 \\ -\left(\dot{m}_L + \dot{m}_{g,s}\right) s_{11} \end{array}\right]$ |

Table 2 Nominal conditions investigated

| $T_{sink}$ $[K]$ | $T_{source}$ $[K]$ | $p_{gc}$ [bar] | $\dot{V}$ [m$^3$/h] | $\Delta T_{pinch}$ [°C] | $\Delta T_{sub}$ [°C] | $\Delta T_{sh}$ [°C] | $\eta_{comp}$ [−] | $\varepsilon_{reg}$ [-] | $T_0$ [K] |
|---|---|---|---|---|---|---|---|---|---|
| 303.2 | 243.2 | 109 | 4.32 | 5.0 | 5.0 | 5.0 | 0.7 | 0.9 | 298 |

## 4. TRANSCRITICAL OPERATION

When the $CO_2$ cycle operates in a transcritical configuration without oil solubility, an optimal oil mass flow fraction can be found which maximizes the system COP for a given set of reservoir temperatures. Figure 4 shows the impact of the oil injection on the individual component irreversibilities and COP as a function of the oil mass flow fraction. This figure shows that the maximum system COP is achieved for the oil mass fraction which minimizes the total irreversibilities of the components impacted by the oil flooding. As the rate of oil injection increases, the pumping/expansion losses of the oil increase, but the heat rejection losses decrease due to the decrease in the size of the superheat horn as the compression process becomes more and more isothermal. The competing trends between the heat rejection and the compression/expansion result in an optimal oil mass fraction dependent on the conditions. While only one temperature source/sink pair is investigated here, the same behavior is seen for other operating conditions. The optimal gas cooler pressure will likely shift as well, but this was not investigated here.

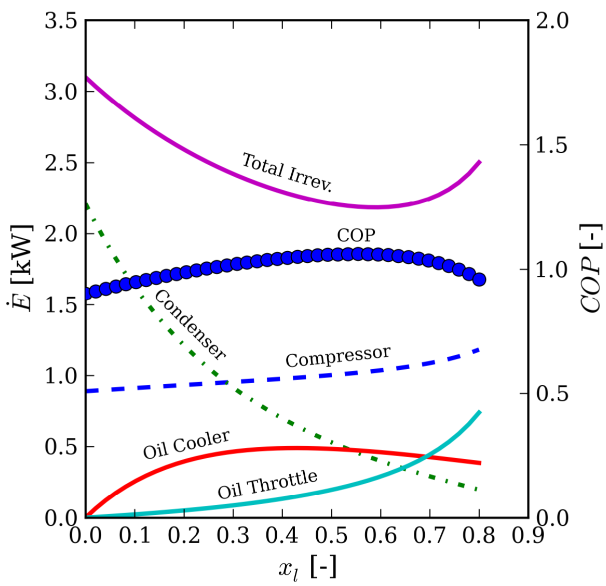
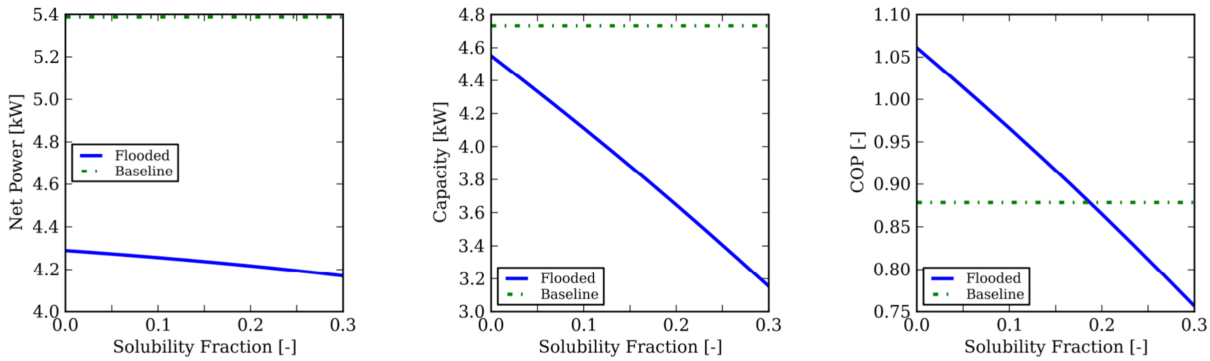



Figure 4 Component irreversibilities and system COP as a function of oil mass fraction without oil solubility

In the next part of the analysis, the oil mass fraction is now set at the same oil mass fraction which optimizes the COP of the system without solubility (55% oil by mass). As the solubility fraction is increased for fixed oil mass flow fraction, the capacity and COP decrease monotonically, and nearly linearly. The non-linearity is due to the fact that the increase in solubility fraction also tends to decrease the inlet temperature to the compressor since the refrigerant which flashes through the oil expansion valve is very cold. As a result, when the cooler oil/flashed-refrigerant mixture mixes with the warm refrigerant exiting the regenerator, the inlet temperature is strongly decreased. This cooling effect also accounts for the slight decrease in net power, since the cooler compressor inlet temperature results in a smaller change in enthalpy over the compressor since the isentropes get steeper as you move towards the saturated vapor curve from the superheated state.

The capacity of the system decreases with more solubility since the refrigerant which is transported through the oil loop still needs to be compressed in the compressor but yields no cooling effect. Thus the impact of oil flashing is to decrease capacity. Since the input power is essentially constant and the capacity falls off steeply, the net effect is a large decrease in the system COP due to refrigerant solubility. If the solubility is increased up to 20% refrigerant by mass, the COP falls back to the COP of the baseline system. Clearly a low-solubility refrigerant must be used for this technology to make sense.




a) Net Power b) Capacity c) System COP

Figure 5 Net Power, Capacity and COP for a fixed oil mass fraction of 55%

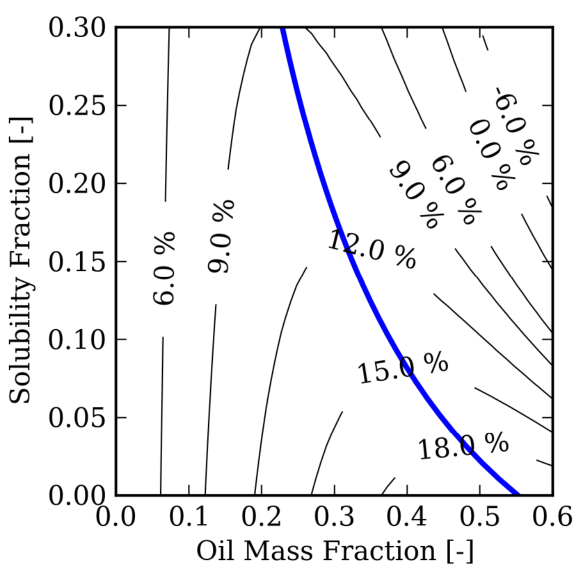



Figure 6 Contour plot of percentage increase in COP with flooding as a function of oil mass fraction and solubility fraction

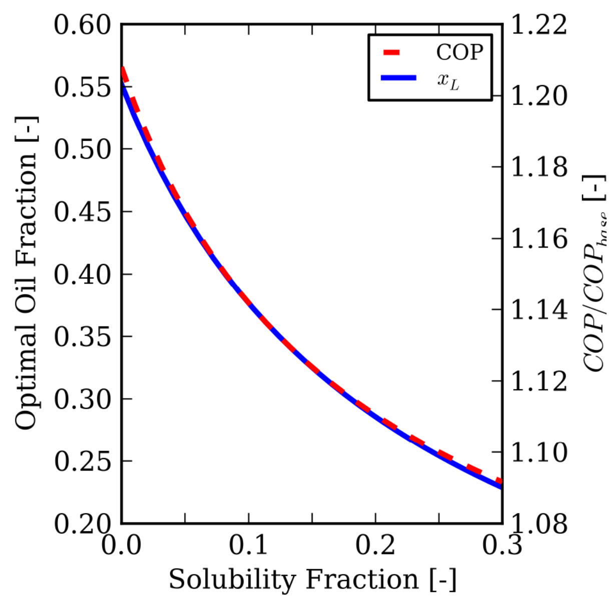



Figure 7 Optimal oil mass fraction and ratio of COP to baseline COP as a function of solubility fraction

As the solubility of the refrigerant in oil increases, the oil mass fraction which optimizes the system COP changes. Figure 6 shows a contour plot of the improvement in system COP from the baseline system as a function of oil mass fraction and solubility fraction. Positive values are systems with better performance than the baseline system. The thick line is the ridge of the surface, or the curve of optimal oil mass fraction for a given solubility fraction. Figure 7 shows the curve for the optimal ridge directly. As the solubility fraction increases, the optimal oil fraction decreases, as does the COP. The optimal oil mass fraction is a tradeoff between more flashing for higher oil mass flow rates and lower component irreversibilities for higher oil mass flow rates.

## CONCLUSIONS

From the analysis presented above, the following conclusions are possible:

- Flooded compression can yield a significant increase in the cooling COP of the vapor compression system, up to 20% for the conditions investigated
- As the solubility of refrigerant in oil increases, the system COP decreases monotonically since the solved refrigerant can no longer provide any cooling effect
- There is an optimal amount of oil flooding which changes depending on the solubility fraction of the refrigerant in oil

## NOMENCLATURE

| Variable | Definition (Units) |
|---|---|
| COP | Coefficient of Performance (-) |
| $\dot{E}$ | Rate of irrev. generation (kW) |
| h | Enthalpy (kJ/kg) |
| $\Delta h$ | Difference in Enthalpy (kJ/kg) |
| $\Delta h_{Reg,max}$ | Max $\Delta h$ in regenerator (kW) |
| $\dot{m}$ | Mass flow rate (kg/s) |
| $s$ | Entropy (kJ/kg-K) |
| $p$ | Pressure (kPa) |
| $T$ | Temperature (K, °C) |
| $\Delta T$ | Difference in temperature (°C) |
| $\dot{Q}$ | Heat transfer rate (kW) |
| $v$ | Specific Volume ($m^3$/kg) |
| $\dot{W}$ | Power (kW) |
| $x$ | Mass flow rate fraction |
| $\varepsilon$ | Effectiveness of Regenerator (-) |
| $\eta$ | Adiabatic Efficiency (-) |

| Subscript | Description |
|---|---|
| 1, 2, 3, … | State Point |
| 1m, 2m, 3m, … | Mixture Property |
| c | Reference |
| comp | Compressor |
| cond | Condenser |
| crit | Critical property |
| evap | Evaporator |
| G | Gas |
| L | Liquid |
| mixer | Mixer |
| oil-cooler | Oil-Cooler |
| pinch | Heat exchanger approach |
| reg | Regenerator |
| sep | Separator |
| sh | Superheat |
| sink | Cold thermal reservoir |
| source | Hot thermal reservoir |
| throttle | Throttle |

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