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Technologies to Improve the Performance of A/C Systems in Hot Climate Regions

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ABSTRACT

Air conditioning contributes significantly to building energy consumption in hot climate regions. In addition to greater cooling requirements in hot climates, cooling equipment efficiency decreases with increasing outdoor temperature. Therefore, it is advantageous to develop improved technologies that can achieve higher efficiency at high ambient conditions.

In this paper, two novel compression technologies are investigated for application in high ambient temperature air conditioning via numerical simulation. These technologies are liquid flooded compression with regeneration and vapor injected compression with economizing. The systems are modeled using the EES software (Klein, 2012) and compared with a baseline conventional vapor-compression cycle that utilizes R410A as the refrigerant. The cycle enhancements are considered for a number of refrigerant alternatives, including R410A, propane (R290), R32, and R1234yf. Parametric studies are conducted for air conditioning design conditions to predict the improvements in coefficient of performance (COP) for both system configurations with the various refrigerants.

The simulation results show that the two novel technologies provide improvements in air conditioner performance and lower compressor discharge temperatures at high ambient temperatures. With respect to compressor discharge temperature, the vapor injection technology is superior to the oil flooding concept for the investigated working fluids. The COP comparisons indicate that oil flooding only improves the system performance when using the refrigerant R1234yf with a 14% increase in COP, whereas the vapor injection leads to significant improvements for all refrigerants with a maximum improvement of 21.5% for the refrigerant R410A.

1. INTRODUCTION AND BACKGROUND

Air conditioning contributes significantly to building energy consumption in hot climate regions. As the usage of air-conditioning (AC) increases worldwide, it is necessary to find ways to reduce its energy consumption for environmental and economic reasons, especially in hot climate regions. However, the system performance decreases with increasing ambient temperature, which makes energy efficient air-conditioning a challenge.

Song (2013) developed a heat pump simulation model with a dual-port vapor injected scroll compressor with a fixed geometry. The dual-port vapor injection reduces the compressor discharge temperatures towards lower outdoor temperatures. The improvements in cycle performance were analyzed and compared to a baseline cycle performance. The results show that vapor injection is suitable for high temperature lift applications and can be applied in cold climate heat pumps. At a condensing temperature of 43.3°C and evaporating temperature of -30°C, the estimated heating COP of the vapor injection cycle was 19% higher than the one of the baseline cycle.

Ramaraj (2012) introduced the oil flooding technology and dual-port vapor injection technology in an R410 cold climate heat with a scroll compressor. It turned out that these two technologies will increase the COP towards lower outdoor temperatures as experienced in cold climate zones. Oil flooding makes it possible for a compressor to operate at very low ambient temperatures where the single stage compressor could not operate due to high discharge temperatures. Both technologies were evaluated via simulation and experimental tests to show their increase in energy efficiency. The results indicate that a similar implementation could be used in hot climate regions.

Bell (2011) reported on liquid flooded compression in scroll compressors. A hermetic R410A air-conditioning compressor was tested with oil injection up to oil mass fractions of approximately 40%. The efficiency of the hermetic R410A compressor that was tested increased with oil injection due to a decrease in heat transfer irreversibility. Bell recommended that a full liquid-flooded system with regeneration should be constructed to better understand the system-level impacts of the addition of the oil and regeneration. It is expected that the performance of the oil-flooded system increases compared to a baseline system without oil flooding. The increase in system coefficient of performance may be greater than 20% depending on operating conditions. With optimization, overall isentropic efficiencies of the compressor of more than 75% are predicted.

Mathison (2011) evaluated advanced compression techniques for vapor compression equipment. The basic cycle model considers two different approaches for economizing with two-stage compression. The first approach uses a flash tank to supply saturated vapor to the compressor between the stages. The second approach to economizing uses an intermediate heat exchanger (IHX) to supply two-phase or vaporized refrigerant to the compressor at the intermediate pressure. For an R-22 cycle operating with an evaporating temperature of -7.2°C, a suction temperature of 7.6°C and a condensing temperature of 48.8°C, the model predicts that a single injection port will increase the COP of the basic vapor compression cycle by up to 12%. Incorporating a second injection port increases the COP of the cycle by 16% over the baseline value without injection.

Hwang and el. (2010) introduced a control strategy for a vapor injection cycle. Their research presents an experimental investigation on the control strategy of the vapor injection system using R410A as its working fluid. A prototype flash tank equipped with a flow visualization window was utilized to monitor and investigate the liquid-vapor separation, which guided the development of the system control strategy. The control of three valves was found to be critical to reliable system operation: the injection port valve, the upper-stage expansion valve and the lower-stage expansion valve.

Wang and el. (2009) investigated a vapor-injected scroll compressor with two cycle options, flash tank configuration (FTC) and internal heat exchanger configuration (IHXC). Both systems were tested for cooling and heating conditions. The results shows that both systems approximately have the same performance results. The improvement in cooling capacity was 14% and COP of 4% at ambient temperature of 46.1°C (Cooling condition) compared to conventional system using conventional scroll compressor. On the other hand, as the ambient temperature decreases (16.7°C to -17.8°C), the heating capacity increase (13% - 33%). The maximum COP is achieved using FTC, and it was 23% higher.

Li and Jin (2004) published the design optimization of an oil-flooded refrigeration system with a single screw compressor. Their research shows that the oil injection can greatly improve the sealing effect. It The study showed that the reduction of clearance, the increase of sealing line, low oil temperatures, and large oil flow rates can greatly reduce the leakage and improve the volumetric efficiency. However, at the same time, the flow resistance loss and the friction power are increased. Thus, the volumetric specific power has to be optimized.

The main objective of this work is to simulate two novel compression technologies for application in high ambient temperature air conditioning. These technologies are liquid flooded compression with regeneration and vapor injected compression with economizing using R410A as the working fluid. These two technologies are modeled using the EES software (Klein, 2012) and compared with a baseline conventional vapor-compression cycle. Analyses are performed to predict the coefficient of performance (COP) for both applications. Parametric studies are conducted to find the optimum operating conditions. Other environmentally friendly refrigerants, including propane (R290) and less flammable refrigerants R32 and R1234yf are explored in the parametric studies.

2. LIQUID FLOODED COMPRESSION WITH REGENERATION SYSTEM

Liquid flooded compression uses oil as a flooding agent, as shown in Figure 1, which is mixed with the superheated suction gas at the internal suction port of the compressor. The oil absorbs part of the heat generated during the compression process (1-2), and is separated from the gaseous refrigerant in the oil separator. The oil is cooled down in the oil cooler (9-10) and then metered through the oil throttle into the suction port. The gaseous refrigerant enters the condenser (3-4), in which it is desuperheated, condensed and subcooled. It enters the regenerator (4-5), where it is further subcooled. The refrigerant is then expanded to evaporation pressure, evaporated and superheated (5-6-7). After the evaporator, the refrigerant is further superheated in the regenerator (7-8) and then enters the compressor suction port (1) together with the liquid oil.

In the oil flooded system, the ratio between the oil and refrigerant have a huge influence on energy efficiency. The aim of this study is to find the optimized oil fractions at different ambient temperature that optimize the COP. Then, the results were compared to the ones of a conventional baseline system for various refrigerants.

Thermodynamic property plots of T-s and P-h diagrams for the oil flooded system including the ones of a baseline cycle are shown in Figures 2 and 3, respectively, for R410A. In can be seen that the enthalpy of the refrigerant increases during the compression process of the baseline system. In comparison, the enthalpy of the refrigerant in the flooded system is relatively constant since the oil absorbs the heat of compression. However, the total enthalpy of oil and gas increases in the flooded compression process. The entropy of the refrigerant decreases during the compression process because of heat transfer to the oil.

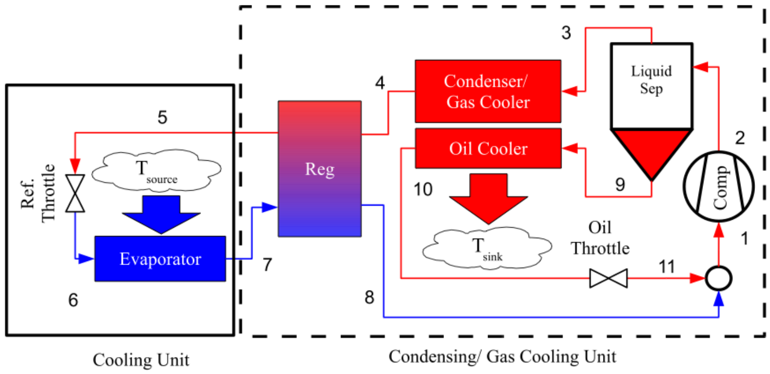

Figure 1: Schematic of flooded vapor compression with regeneration system (Bell, 2011)

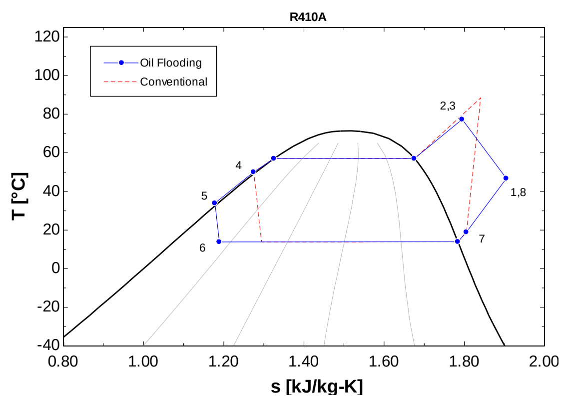

Figure 2: T-s diagram for oil flooded compression cycle using R410A (x_l = 60%)

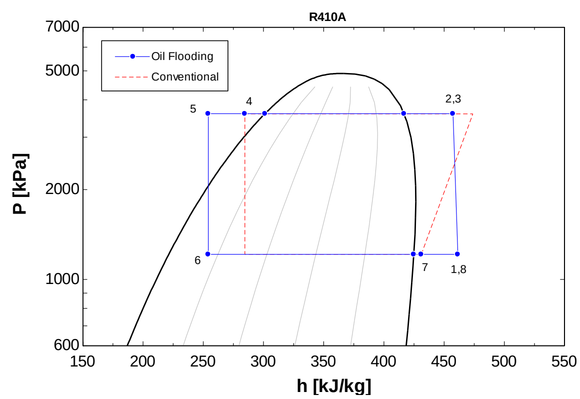

Figure 3: P-h diagram for oil flooded compression cycle using R410A (x_l = 60%)

2.1 Modeling Oil Flooded Compression System

Engineering Equation Solver (Klein, 2012) is used in this work to model the liquid flooded compression cycle. The thermodynamic modeling of flooded compression cycles (Bell et al., 2008) is similar to that of standard, one-fluid, vapor compression systems. The main differences arise from considering the oil and its properties during the compression process and in the oil loop. Several assumptions are applied for the mixture properties. It is assumed that the oil and refrigerant are in thermal and mechanical equilibrium and thus, both the oil and refrigerant phases are at the same temperature and pressure. For the cycle analysis, the necessary thermodynamic mixture properties are the mixture enthalpy and mixture entropy as functions of temperature and pressure. Thus, the mixture enthalpy and mixture entropy can be expressed as

$$h_m = x_l h_l + (1 - x_l) h_g \tag{1}$$

$$s_m = x_l s_l + (1 - x_l) s_g \tag{2}$$

which is simply an oil-mass-flow-fraction weighted average of the properties of the separated oil and gas phases. The oil mass fraction x_l is given as the ratio of mass flow rates of oil to total mass flow rate, defined by

$$x_l = \frac{\dot{m}_l}{\dot{m}_l + \dot{m}_g} \tag{3}$$

and the gas mass fraction is given by

$$x_g = 1 - x_l \tag{4}$$

Simple models are presented in Table 1 for each of the cycle component.

Table 1: Equations used to model oil compression system

Compressor	Condenser	Oil Separator	Oil Cooler / Expansion
$h_1 = h_m(T_1, P_1, x_{l,1})$ $s_1 = s_m(T_1, P_1, x_{l,1})$ $h_{2s} = h_m(P_2, s_1, x_{l,1})$ $\eta_{comp} = \frac{h_{2s} - h_1}{h_2 - h_1}$ $s_2 = s_m(P_2, h_2, x_{l,2})$ $\dot{W}_{comp} = (\dot{m}_l + \dot{m}_g)(h_{2m} - h_{1m})$	$T_4 = T_{sink} + \Delta T_{pinch}$ $h_4 = h_m(T_4, P_4, x_l = 0)$ $s = s_m(T_4, P_4, x_l = 0)$ $\dot{Q}_{cond} = \dot{m}_{g,v}(h_4 - h_3)$	$T_2 = T_3 = T_9$ $P_2 = P_3 = P_9$ $x_{g,s} = \frac{\dot{m}_{g,s}}{\dot{m}_{g,s} + \dot{m}_l}$	$T_{10} = T_{sink} + \Delta T_{pinch}$ $\dot{Q}_{oil\ cooler,l} = \dot{m}_l C_{p,l}(T_9 - T_{10})$ $\dot{Q}_{oil\ cooler,g} = \dot{m}_{g,s}(h_{10,g} - h_{9,g})$

Expansion Valve / Evaporator	Regenerator	Mixing
$T_{evap} = T_{source} - \Delta T_{pinch} - \Delta T_{sh}$ $h_7 = h_m(T_{evap} + \Delta T_{sh}, P_{evap}, x_l = 0)$ $\dot{Q}_{evap} = \dot{m}_{g,v}(h_7 - h_6)$	$\Delta h_{Reg,max} = min\begin{cases} h_4 - h(T_7, P_4) \\ h(T_4, P_7) - h_7 \end{cases}$ $\varepsilon_{Reg} = \frac{\Delta h_{Reg}}{\Delta h_{Reg,max}}$ $h_5 = h_4 - \Delta h_{Reg}$ $h_8 = h_7 + \Delta h_{Reg}$	$(\dot{m}_l + \dot{m}_{g,s})h_{11} + \dot{m}_{g,v}h_8 - (\dot{m}_l + \dot{m}_{g,v} + \dot{m}_{g,s})h_1 = 0$

The system efficiency must be defined in order to characterize the flooded and non-flooded systems. The cycle efficiency or Coefficient of Performance (COP) is defined as follows:

$$\mathrm{COP} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} \tag{5}$$

The only free parameter affecting the compression power in the oil flooding cycle is the oil mass flow fraction. The oil flow rate impacts five irreversibility terms: irreversibilities in the throttle valve, compressor, condenser, oil cooler and the mixer. In this study, the optimal oil mass flow fraction was found for each operating condition that maximizes the system cooling COP for air-conditioner using golden section method performed in EES (Klein, 2012).

2.2 Design Conditions and Model Assumptions

Design conditions for the A/C system are a cooling capacity of 10 kW (3 tons of cooling) at a design indoor temperature of 23.88°C (75°F). This is a typical assumption used as a design condition for hot climates. The performance of the system is estimated for various ambient temperatures and different working fluids.

R410A is used for the system analysis because most of the air conditioning applications on the market today use R410A as the refrigerant. Other environmentally friendly refrigerants, including propane (R290) and the less flammable refrigerants R32 and R1234yf are also investigated. The analysis is performed for fixed heat source (cool space) and heat sink (ambient) temperatures as well as superheat, subcooling and pinch point temperature differences. Some general assumptions include:

- All state points of the cycle are subcritical.
- The throttling device is assumed to operate adiabatically to maintain a specified superheat exiting the evaporator.
- Pressure drops in the heat exchangers are neglected.
- ΔT_{pinch} = 5°C, ΔT_{sh} = 5°C and ΔT_{subl} = 7°C for the outlet of the heat exchangers.
- For air conditioning mode, the sink temperature (T_H) is the ambient temperature ranging between 20°C to 55°C and the source temperature (T_L) is kept as a constant 23.88°C.
- Compressor isentropic efficiency is assumed to be 70%.
- Saturation temperatures T_{cond} and T_{evap} are calculated as follows:

$$T_{evap} = T_L - \Delta T_{sh} - \Delta T_{pinch} \tag{6}$$

$$T_{cond} = T_H + \Delta T_{sub} + \Delta T_{pinch} \tag{7}$$

$$T_{cond,out} = T_H + \Delta T_{pinch} \tag{8}$$

$$T_{evap,out} = T_L - \Delta T_{pinch} \tag{9}$$

POE Oil is used as a flooding agent for liquid flooded compression because of its wide temperature miscibility range, it is hydrolytically excellent, has oxidation and chemical stability, and good lubricity properties. A modified specific heat correlation from Conde (1996) & Lee and Kesler (1976) is used to determine the properties of POE oil:

$$C_p\left(\frac{kJ}{kg.K}\right) = -1.4271 + 0.00832T(K) \quad (10)$$

For flooding purpose, it is assumed that the refrigerant and oil are in mechanical and thermal equilibrium and the solubility of refrigerant in oil is only considered in the oil separator. In the separator it is assumed that all the liquid is fully separated and passes through in the liquid phase. There is no pressure drop, but the fictitious change in solubility results in some of the refrigerant going with the oil in solution rather than staying in the vapor phase. When some amount of refrigerant is solved in oil, the cooling capacity decreases. For simplicity, it is assumed that no refrigerant is dissolve in the oil. At each ambient temperature, the optimal oil mass flow fraction is found which optimizes the system cooling COP. The regenerator is a critical component of the flooded vapor compression system and the regenerator effectiveness is assumed to be 90%.

3. VAPOR INJECTED COMPRESSION WITH ECONOMIZING SYSTEM

Another method to increase the performance of an air conditioning system is the vapor injection with economizing technology. Figure 4 shows a schematic of the vapor injection cycle with flash tank economization for a single injection point and intermediate pressure. With refrigerant injection, a stream of refrigerant exiting a flash tank or intermediate heat exchanger (IHX) economizing is diverted from the evaporator to provide cooling during the compression process. IHX economizing would require a 100% effective IHX to match the performance of flash tank economization (Mathison, 2011). Since an IHX will have an effectiveness of less than 100% in practice, flash tank economization is identified as the most promising cycle modification for performance improvements. A flash tank operating at an intermediate pressure separates the liquid and vapor phases of the refrigerant. A part of the refrigerant from the flash tank is drawn off and mixed with refrigerant at an intermediate stage within the compressor to cool the refrigerant during the compression process and thus, reduce the compression work. The reduction in compression work results from lowering the superheat of the refrigerant in the compressor, which simultaneously reduces the isentropic compression work and increases the mass flow rate through the compressor. Since the compression process more closely follows the saturated vapor line, the desuperheating losses in the condenser also decrease. The saturated liquid from the flash tank is expanded from the intermediate pressures and sent to evaporator. Since the expansion process tends to occur much more closely to the saturated liquid line, the expansion losses also decrease.

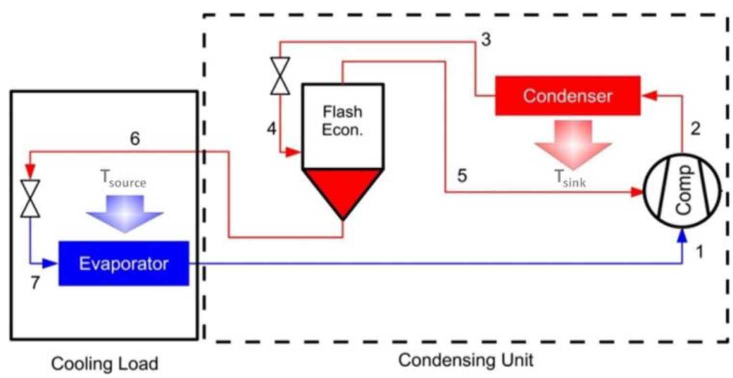

Figure 4: Schematic of a flash tank vapor injection cycle (Ramaraj, 2012)

The refrigerant drawn from the flash tank could be either two-phase liquid or saturated vapor. Two-phase injection provides greater performance improvements than the saturated vapor injection because the two-phase refrigerant delivers additional cooling to the compressor. However, the improvement in COP is not significant and there is a challenge to control the quality for the two-phase system. Also, the cost for the two-phase system is much higher than the saturated vapor injection system. Although the two-phase injection provides a somewhat greater improvement in COP, only saturated vapor injection is considered in this study.

Extensive work has been done on the vapor injection system heat pump. Previous research indicates that the COP increases with an increase in the number of refrigerant injection ports (Mathison et al., 2011). For R-410A heat pump cycles, most of the benefits associated with continuous injection can be achieved with 2 to 4 injection ports. Using three ports to inject saturated vapor improves the cycle COP by 32% which is 70% of the maximum improvement in COP that can be achieved with economization when the cycle operates with an evaporating temperature of -20°C, a condensing temperature of 50°C, and a compressor isentropic efficiency of 70%. Usually the system is chosen for a multi-port vapor injection system on the trade-off between cost and COP. In typical air conditioning operating conditions, the performance does not improve much between the single port and dual port system because of the limited temperature lift. Figure 5 shows coefficient of performance for the single port and dual port system when the high side temperature changes from 25°C to 50°C. The improvement is less than 1%. Since it is more economical, only a single port air conditioner vapor injection system is considered here.

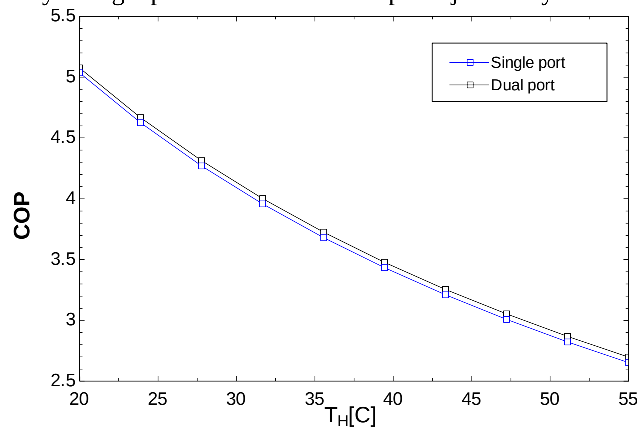

Figure 5: Comparison of COP between single port and Dual-port vapor injection system in A/C mode

3.1 Modeling of Vapor Injection System

Figure 6 shows the P-h diagram for the single port vapor injection technology. A flash tank separator is installed between the first and second expansion valves. The refrigerant that is evaporated in the first expansion valve enters the compressor during the compression process, which is modeled as a two-stage compressor. The vapor from the evaporator flows into the compressor at point 1, and is compressed to point 4 by the scroll compressor. The horizontal line between point 2 and point 3 shows the influence of the saturated vapor refrigerant coming from the flash tank. The liquid at the outlet of the flash tank separator flows into the second expansion valve, then into evaporator, and back to compressor.

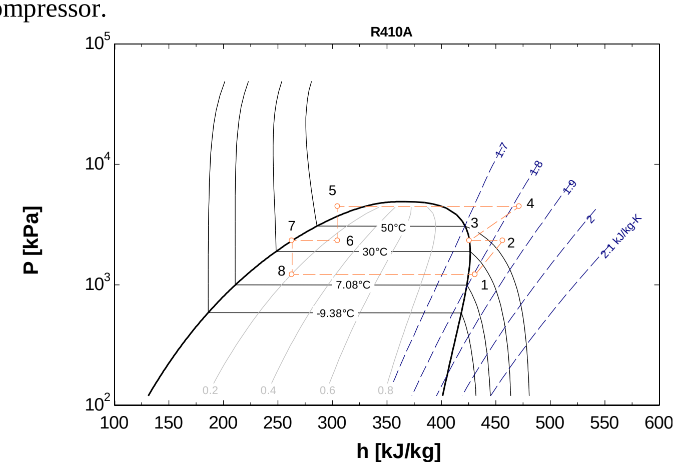

Figure 6: P-h diagram for R410A with single-port flash tank vapor injection system

The condensing and evaporating temperatures are the same as the ones in the oil flooded system, with ΔT_{pinch} standing for the temperature difference between the heat exchanger temperature and the ambient temperature, and T_{sh} and T_{sub} standing for the superheated and subcooling temperature difference. With proper assumptions, all the thermodynamics characteristics are obtained using EES (Klein, 2012).

A mass balance on the flash tank separation process relates the mass flow rate in the main refrigerant line and in the injection line. In the single port injection system, the quality is determined by the condenser temperature and pressure as an isenthalpic process in the expansion valve. The mass flow rates through these lines are defined as a fraction of the total mass flow rate through the condenser, where $\dot{r}_{comp}$ is the fraction of the total flow rate that enters the compressor at pressure p_{inj}, and $\dot{r}_{inj}$ is the fraction of the total mass flow rate through the injection line at pressure p_{inj}.

$$\dot{r}_{comp} = \frac{\dot{m}_{comp}}{\dot{m}_{cond}} \tag{11}$$

$$\dot{r}_{inj} = \frac{\dot{m}_{inj}}{\dot{m}_{cond}} \tag{12}$$

While both of the mass fractions are unknown, combining the mass balance with energy balance of the whole system to calculate the mass fraction and enthalpy at each working state.

In the compression process, inlet and outlet temperatures are determined by the working conditions, and the working pressure is found by assuming a saturated refrigerant. After the isobaric process in the condenser, all the enthalpies are calculated using proper pressure assumptions.

$$\dot{r}_{comp}[i] = \dot{r}_{comp}[i+1](\frac{h_6[i]-h_3[i]}{h_7[i]-h_2[i]}) \tag{13}$$

$$\dot{r}_{inj} = \dot{r}_{comp}[i+1] - \dot{r}_{comp}[i] \tag{14}$$

$$h_9[i] = \frac{\dot{r}_{comp}[i+1]h_6[i]-\dot{r}_{comp}[i]}{\dot{r}_{inj}} \tag{15}$$

The isentropic efficiencies are used considering that the injected refrigerant is compressed parallel to the suction refrigerant. When calculating isentropic work, the suction refrigerant is assumed to be calculated from point 1 to 4s in the high-pressure horizontal line isentropically, and the injected refrigerant is compressed from point 3 to point 3s also in the high-pressure horizontal line isentropically. The calculation of isentropic efficiency of the vapor injection compressor is shown in the following equations to calculate the discharge temperature and compressor outlet enthalpy.

$$\eta_{isen} = \frac{\dot{W}_{isen}}{\dot{W}_{comp}} \tag{16}$$

$$\dot{W}_{comp} = \dot{m}_{comp}(h_4 - h_1) + \dot{m}_{inj}(h_4 - h_3) \tag{17}$$

$$\dot{W}_{isen} = \dot{m}_{comp}(h_{4s} - h_1) + \dot{m}_{inj}(h_{3s} - h_3) \tag{18}$$

With these constraints, the only two independent parameters are high-side temperature and low-side temperature. By assuming a constant isentropic efficiency and volumetric efficiency, the system could be modeled to find out the thermodynamic performance.

3.2 Design Conditions and Model Assumptions

To compare vapor injection and oil flooded technologies with the base-line cycle, the design conditions are assumed the same as the oil flooded model for A/C mode: having a cooling capacity of 3 tons at a design outdoor temperature of 23.88°C. The performance is estimated for the same range of sink temperatures for different refrigerants.

The vapor injection model uses R410A as the refrigerant because of its common use in air condition applications. Other environmentally friendly refrigerants, such as R290, R1234yf, and R32, are also considered as a comparative study in this analysis. The analysis is conducted with the same heat source and heat sink temperatures and the same superheat, subcooling and pinch temperature as the oil flooded analysis. The ambient temperature range is set from 20°C to 55°C to determine the percentage of COP variation and compare the results for various refrigerants. Some proper assumptions are made to simplify the vapor injection system. The assumptions include:

- The injection pressures were selected to result in equal pressure ratios between the injection points.
- Compressor isentropic efficiency is fixed at 70%.
- Compressor volumetric efficiency is fixed at 100%.
- All the state points are subcritical.
- Pressure drop in heat exchangers is neglected.
- Process in expansion valve is isenthalpic.
- ΔT_{pinch} = 5°C, ΔT_{sh} = 5°C and ΔT_{sub} = 7°C for the outlet of the heat exchangers.
- Fan work is neglected.
- For air conditioning mode, the sink temperature (T_H) is the ambient temperature varies from 20°C to 55°C and the source temperature (T_L) is fixed at 23.88°C.
- The cooling capacity is 3 tons of cooling.

In the flash tank, the vapor and liquid is assumed to be totally separated. The liquid flows into the evaporator and the vapor enters the compressor to decrease the superheat process in the compressor. The quality is determined by the isenthalpic process in the first expansion valve.

4. RESULTS AND DISCUSSION

Figure 7 shows the compressor discharge temperature with respect to ambient temperature for the vapor injected, oil flooded and conventional systems using R410A, R1234yf, R290, and R32. As the ambient temperature increases, the reduction in discharge temperature becomes more obvious. In the oil flooded system, the discharge temperature is higher than the one of the conventional system. A proper explanation for this phenomenon is that in oil flooded systems there is a regenerator causing a higher refrigerant temperature at the compressor inlet. This is true for all investigated refrigerants except R32. In the oil flooded system with R32 as the working fluid, there is improvement reduction in the discharge temperature of 10°C at higher ambient conditions. Although R32 in oil flooded system shows better enhancement, the discharge temperature in this case, 110°C at T_H = 55°C, is relatively higher than the one for the other refrigerants. The other refrigerants, R410A, R123yf and R290, have discharge temperatures of less than 110°C when operated at higher ambient temperature. Generally, the compressor discharge temperature in the oil flooded system can be improved with more oil mass injected into the compressor.
On the other hand, the compressor discharge temperature in the vapor injected system shows more improvement compared to the oil flooded and conventional systems. As R410A and R32 are the working fluids, a significant improvement in the discharge temperature can be noticed at higher ambient temperature conditions. Although R290 and R1234yf have relatively small improvement in discharge temperature, their discharge temperatures are significantly lower (less than or equal to 80°C) than the ones of R410A and R32.

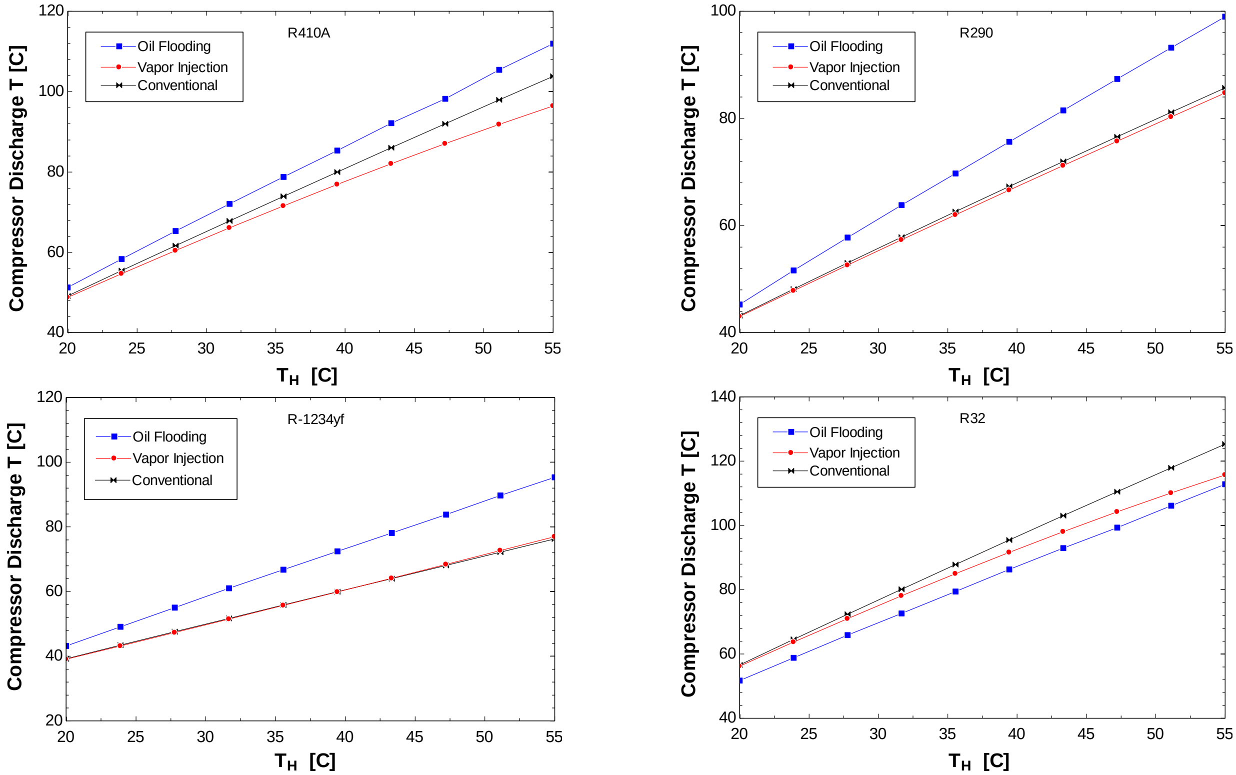

Figure 7: Compressor discharge temperature comparison between oil-flooded, vapor injection and conventional systems using R410A, R1234yf, R290, and R32

The performance improvements with respect to ambient temperature for the vapor injected, oil flooded and conventional systems using R410A, R1234yf, R290, and R32 are illustrated in Figure 8. The vapor injection technology shows a significant performance improvement compared to oil flooded system at higher ambient temperature conditions. The improvement are 21.5%, 6%, 16.5% and 16.5% with the refrigerants R410A, R1234yf, R290 and R32, respectively. On the other hand, relatively small improvements in COP are obtained with the oil flooded system with the different types of refrigerants. The improvement varies between 3 to 7% for the investigated refrigerants except R1234yf. R1234yf shows a significant improvement in COP of 14% with the oil flooded system.

This case would be appropriate for experimentally investigating oil flooded technology with R1234yf for hot climate conditions. Also, Figure 8 indicates that for ambient temperatures of less than 35°C, no improvement in the performance of the oil flooded system can be observed. The reason for this is the high compressor inlet temperature.

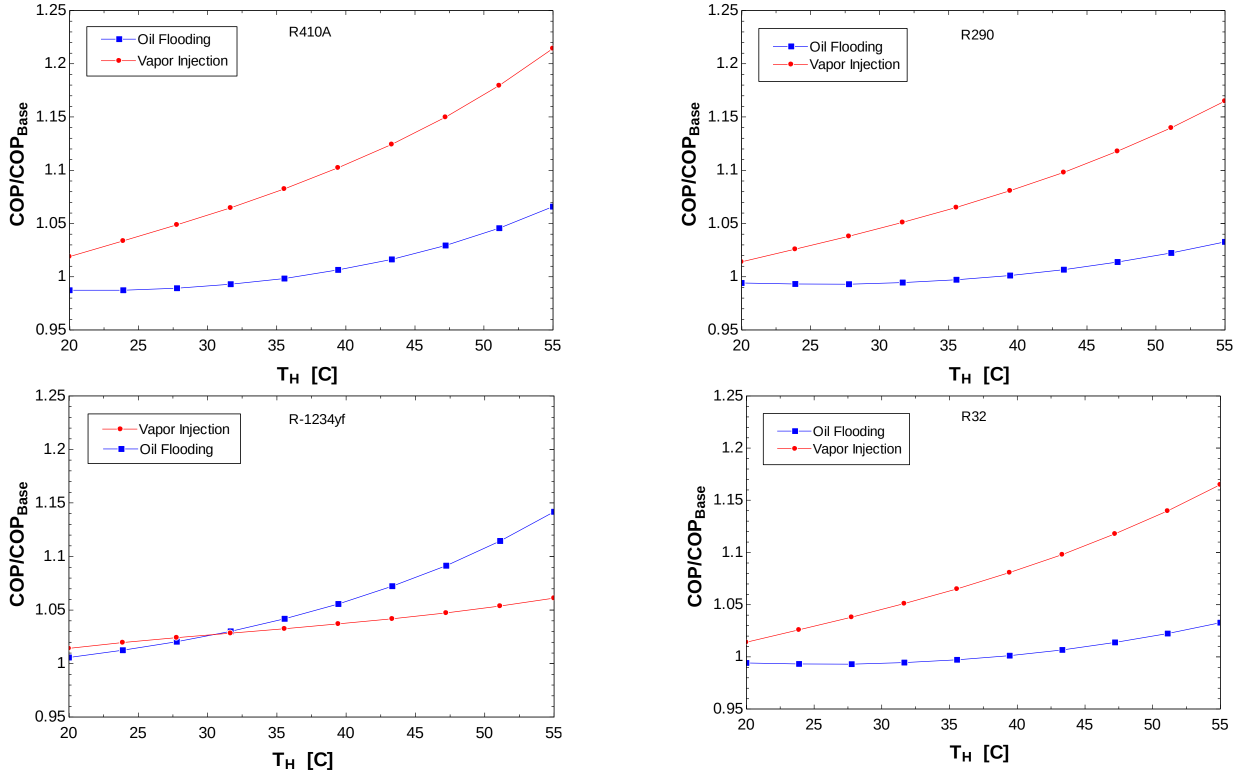

Figure 8: Performance improvement of VI and Oil flooded systems using R410A, R1234yf, R290, and R32

5. CONCLUSIONS

In this work, models for the oil flooding with regeneration technology and vapor injection with economizing technology are implemented in EES (Klein, 2012) to simulate the coefficient of performance and compressor discharge temperature. Analyses are performed for air conditioning working conditions using four different refrigerants, including R410A, R290, R32 and R1234yf. Comparisons are conducted showing compressor discharge temperatures and COPs of the oil flooded and vapor injected systems against a conventional baseline system to explore the suitability of the technology in air conditioning mode. The comparisons are performed under specific design conditions and assumptions.

With respect to compressor discharge temperature, the vapor injection technology shows significant improvement with the investigated refrigerants except for R32. The discharge temperature in vapor injection is approximately 15°C lower than the one of the conventional system using R410A and 10°C lower for R32. However, for R290 and R1234yf, there are only relatively small improvements in compressor discharge temperature. On the other hand, the compressor discharge temperature of the oil flooded system is higher than the conventional system because of the higher compressor suction temperature for all refrigerants except R32. With the increase in oil injection, the compressor discharge temperature can be reduced in contrast to the COP.

With respect to COP, the vapor injection technology shows significant improvements for the investigated refrigerants except for R1234yf. The percentage improvement in COP at high ambient temperature of T_H = 55°C varies from 15% to 20% for R410A, R290 and R32. On the other hand, the oil flooded system does not perform well for these three refrigerants with respect to COP improvement. The maximum increase is approximately 5%, and the COP is even lower than the conventional system when the ambient temperature is lower than 35°C. However, the oil flooding technology shows significant performance improvement for R1234yf. The enhancement of COP is approximately 15% while the vapor injection technology shows no improvement with R1234yf.

In summary, the simulation results shows that vapor injection technology has better performance with R410A, while R1234yf is more suitable in an oil flooded system for high temperature air conditioning mode. In the next step, it is suggested to experimentally investigate the oil flooding technology using R1234yf and the vapor injection technology using R410A. For this study, the injected refrigerant is assumed to be saturated vapor. The vapor injection model could be improved by controlling the quality of the injected refrigerant. In the future, an improved model could be built up by adding a control valve at the outlet of the flash tank to control the injection quality to enhance both compressor discharge temperature and COP.

NOMENCLATURE

h	enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/sec)
s	entropy	(kJ/kg-k)
P	pressure	(kPa)
$\dot{W}$	power	(kW)
C_p	specific heat	(kJ/kg-K)
T	temperature	(°C)
Q	cooling capacity	(kW)
x	mass fraction	(–)
η	efficiency	(–)
$\dot{r}$	mass fraction	(–)
COP	coefficient of performance	(–)
IHX	intermediate heat exchanger	(–)

Subscript

m	mixture
l	oil
g	refrigerant
g,v	refrigerant in vapor phase
g,s	refrigerant in liquid phase
comp	compressor
evap	evaporator
cond	condenser
sh	superheat
sub	subcooled
Reg	regenerator
L	source
H	sink
inj	injection
isen	isentropic
1,2,…	state number
1s,2s,…	isentropic state number
1m,2m,…	mixture state number

REFERENCES

Bell, Ian, "Theoretical and Experimental Analysis of Liquid Flooded Compression in Scroll Compressors" (2011). Publications of the Ray W. Herrick Laboratories. Paper 2.

Mathison, Margaret M., Modeling and Evaluation of Advanced Compression Techniques for Vapor Compression Equipment (2011).

Song, Yuanpei, Modeling and Experimental Validation of a Multi-port Vapor Injected Scroll Compressor (2013).

Ramaraj, Sugirdhalakshmi. Vapor Compression Cycle Enhancements for Cold Climate Heat Pumps (2012).

Bell, Ian H., Groll, Eckhard A., Braun, James E., and Horton, W. Travis, "Impact of Oil Solubility and Refrigerant Flashing on the Performance of Transcritical CO2 Vapor Compression Systems with Oil Flooding and Regeneration" (2010). International Refrigeration and Air Conditioning Conference.Paper 1025.

Bell, Ian H., Groll, Eckhard A., and Braun, James E., Performance of Vapor Compression Systems with compressor Oil Flooding and Regeneration. (2010). Publications of the Ray W. Herrick Laboratories. Paper 4.

Hwang, Yunho, Xu, Xing, Radermacher, Reinhard, and Pham, Hung, "Control Strategy of Vapor Injection Cycle" (2010).International Refrigeration and Air Conditioning Conference.Paper 1066.

Wang, Xudong, Hwang, Yunho, and Radermacher, Reinhard, "Two-stage heat pump system with vapor-injected scroll compressor using R410A as a refrigerant" (2009). International Journal of Refrigeration 32 (2009) 1442–1451.

Li, Hong Qi and Jin, Li Wen, "Design Optimization of an Oil-Flooded Refrigeration Single Screw Compressor" (2004).InternationalCompressor Engineering Conference. Paper 1715.

Jang, Yonghee; Lee, Eungyeol; Chin, Simon; and Ha, Samchul, "Effects of Flash and Vapor Injectionon the Air-to-Air Heat PumpSystem" (2010). International Refrigeration and Air Conditioning Conference.Paper 1138.

Chukanova, Ekaterina; Stosic, Nikola; Kovacevic, Ahmed; and Dhunput, Ashvin, "Investigation of Start Up Process in Oil FloodedTwin Screw Compressors" (2012). International Compressor Engineering Conference.Paper 2048.

Klein, S. A. (2012). Engineering equation solver, academic commercial version 9.434-3d. F-Chart Software, LLC. Madison, WI.