2022

Quantitative Comparison of the Performance of Vapor Compression Cycles with Various Means of Compressor Flooding

Amjid Khan

Craig R. Bradshaw

Quantitative Comparison of the Performance of Vapor Compression Cycles with Various Means of Compressor Flooding

Amjid Khan[1]*, Craig R. Bradshaw[2]

[1,2]Center for Integrated Building Systems, Oklahoma State University,
Stillwater, Oklahoma, US
[1]amjid.khan@okstate.edu, [2]craig.bradshaw@okstate.edu

* Corresponding Author

ABSTRACT

Increasing the Coefficient of performance (COP) of vapor compression cycles has been a major focus for decades. Significant improvements can be brought by modifying multistage compressor to flooded single-stage compressor in order to approach isothermal compression, which reduces compressor work and improves COP. This work presents the comparative analysis of the cycle performance of a 3 stage compression system (3SCS), multiple refrigerant injection system (MRIS) and an oil flooded system (OFS). Thermodynamic models of each of these systems using ideal mixing models of refrigerants with the flooding agents is presented. Results show that in an MRIS, an increase in the number of injection ports significantly increases the COP of the cycle between 18% and 51% for air conditioning and refrigeration applications. The OFS shows that increasing the oil mass flow rate increases the COP significantly above the baseline cycle, benefiting higher temperature lift applications. At evaporating temperature of -40°C, relative COP for MRIS with 3 injection points is 3.5% higher than 3SCS and 5.88% higher than OFS. Further, quantification of the influence of the non-ideal mixing model showed that dropping mixing effectiveness from 100% to 80% results in 21% degradation of COP.

1. INTRODUCTION

As vapor compression cycles are widely used in heating and cooling applications, even slight improvements in vapor compression equipment efficiency can have a considerable influence on total energy consumption. Modifications to the fundamental cycle design provide a way to improve performance over a traditional cycle. Two typical modifications include intercooling and economizing that both attempt to minimize compressor power consumption and, in some situations, improve cooling capacity (Xu *et al*. 2016).

When there is high temperature lifts, refrigeration and heat pump units using a single stage vapor compression system suffer from reduced efficiency. As a result of this, the system has several limitations, as discussed below from Dalkiliç (2011):

- High compressor discharge temperature causes thermal instability in the lubricating oil.
- Volumetric efficiency drops dramatically, reducing cooling capacity.
- Low COP, as compressor efficiency declines at high temperatures.

The most generally utilized approach to overcome the restrictions caused by higher temperature and pressure lifts, as well as the limitations associated with single stage compression, is multistage compression. This technique comprises of multiple compressors. Multiple stage compression has been investigated both numerically and experimentally by several researchers in order to illustrate the technique's potential for reducing cycle work and increasing COP. Thermodynamic analysis and performance simulation showed that the COP for two-stage flash intercooling transcritical CO2 cycle improves slightly, as the intermediate pressure drops below the geometric mean of the evaporator and gas cooler pressures (Agrawal & Bhattacharyya, 2007). Experimental evaluation of a two-stage refrigeration cycle for a wide range of evaporating and condensing temperatures was performed, and it was observed that a two-stage refrigeration cycle with a subcooler provides higher cooling capacity and the best COP at higher condensing and evaporating pressures (Cabello *et al*. 2010).

Jiang *et al*. (2015) developed a numerical model for the two-stage compression cycles with the flash intercooler and subcooler. It was found from the analysis, when the heat exchanger subcooling parameter "ε" is greater than 0.7, the COP_h for the subcooler cycle is higher. It was also determined that larger the refrigerant molecules' specific heat capacity, the greater the benefit of interstage cooling. The two-stage centrifugal compressor cycle was studied numerically. The simulation findings showed that COP is only a valid criterion for comparing cycles with the same evaporation and condensation temperatures and cooling power requirements. It's also worth noting that the two-stage compression cycle outperforms the single-stage compression cycle (Röyttä *et al.*, 2009).

Bertsch & Groll (2008) simulated, designed, and built an R410A-based air-source two-stage heat pump. They verified that the heat pump could work at ambient temperatures as low as -30°C to 10°C and supply water temperatures as high as 50°C in heating mode. When compared to the single-stage mode, the two-stage mode nearly doubled the heating capacity at the same ambient temperature. In the two-stage mode, the compressor discharge temperatures remained below 105°C, whereas single-stage mode lingered beyond 118°C. According to the literature, multistage compression surpasses single-stage compression, but the expense has confined the scope of these modified cycles to large-scale applications.

Multiple refrigerant injection systems and oil flooding systems for the compression process have introduced cost effective novel cycle modification alternatives in order to avoid the high expense of multistage compression cycles. These systems have several advantages, the most important are discussed by Xu *et al.* (2011) as follows:

- Controlling the injected refrigerant mass flow rate can change the system capacity, allowing for some energy savings by avoiding the compressor's intermittent operation.
- The discharge temperature is also low for flooding compressors.

Copeland Corporation has patented a scroll compressor with one refrigerant injection port and after that scroll compressor with vapor injection is one of the most utilized compressors in vapor injection heat pump systems. Using a scroll compressor vapor injection with an internal heat exchanger (economizer), Ma *et al.*(2003) conducted an experimental research of air-source heat pumps for cold climates. The prototype was able to operate smoothly at temperatures as low as -15°C, with enhanced heating capacity and COP, and a stable discharge temperature of less than 130°C. Ma & Zhao (2008) compared the heating performance of a heat pump with a flash tank and scroll compressor vapor injection to a system with an economizer cycle using R-22. At air temperatures of 45°C in the condenser and -25°C in the evaporator, the flash tank cycle's heating capacity and COP were 10.5% and 4.3% greater, respectively, than the economizer cycle.

Feng *et al.* (2009) investigated a heat pump water heater with a vapor injection mechanism and an R22/R600a mixture. They observed that when vapor injection was employed, the heating capacity and COP of the system with mixed refrigerant of R22/R600a were 30% and 32% greater, respectively, than when R22 was used as the refrigerant. Wang *et al.* (2009) examined the performance of a heat pump system with scroll compressor vapor injection and cycle selections of both flash tank and economizer configurations using R410A refrigerant. When compared to a typical heat pump system, they found a cooling capacity gain of roughly 14% with a 4% COP improvement at an ambient temperature of 46.1°C in cooling mode, and a heating capacity gain of around 30% with a 20% COP gain at an ambient temperature of -17.8°C in heating mode.

Mathison *et al.* (2010) developed a basic EES model to demonstrate the effect of multiple refrigerant injection into the compressor, and concluded that at high temperature lifts, R-404A shows a COP improvement of 51% at an evaporating temperature of -30°C and a condensing temperature of 40°C for supermarket refrigeration cycles. R410A exhibited an 18% increase in COP, while R134a showed a 22% increase in COP.

On the other hand, Liquid injection or oil flooding cycles, have been designed to achieve quasi-isothermal compression, which reduces compressor work and improves the systems COP. The efficacy of compressor flooding systems increases at high temperature lift applications such as supermarkets. To achieve quasi-isothermal compression, a considerable amount of high specific heat fluid is added to the refrigerant flow to absorb the heat of compression. In practice, lubricity dictates that the fluid be liquid oil, but other fluids are viable. Hugenroth *et al.* (2008) has investigated liquid flooding extensively in order to approximate the Ericsson cycle and has also shown that water is the optimum flooding liquid because of its low specific volume and high specific heat. The low specific volume is advantageous since the liquid must be pushed to the compressor discharge pressure together with the refrigerant, and a low specific volume reduces the liquid specific compression effort. Furthermore, a high liquid specific heat means that less liquid is needed to absorb the same amount of heat from compression.

Coney *et al*. (2002) studied the influence of the design variables on the performance of the oil injected single-stage screw compressor and observed that clearance, oil injection temperature, and oil quantity of the leaking channel has the greatest influence on the performance of the compressor. Bell *et al.* (2011) studied the addition of the compressor flooding with regeneration in vapor compression systems for achieving isothermal compression to help system increase the efficiency for large temperature lift applications.

Patil *et al*. (2020) experimentally studied the water spray injection in a liquid piston cylinder for a pressure ratio of 2.5 in order to assess the effectiveness of spray injection. It was observed that discharge temperature reduction increases with the increase of injected pressure of the water spray from 10 psi to 70 psi. The pressure-volume curves were observed to achieve quasi-isothermal curves with the water spray pressure above 30 psi and at water spray pressure of 70 psi showed 21% improvement in the isothermal efficiency of the compressor.

To the best of our knowledge, there is no comprehensive systematic comparison of these three technologies: 3SCS, MRIS, and OFS, which covers the whole range in which they are working. We also noticed that there is no quantitative study available on the influence of the non-ideal mixing model in terms of mixing effectiveness of the compressor on the resulting cycle performance. For this reason, we developed a comparative study in terms of Coefficient of Performance (COP), compressor work, and mixing effectiveness (OFS) of the compressor. The comparative study was performed with high pressure ratio and high temperature differences between evaporating and condensing temperatures.

2. METHODOLOGY

This study compares the compressor performance using three different systems 1) a three-stage compression (3SCS) 2) multiple refrigerant injection (MRIS), and 3) oil flooded system (OFS). This study was analyzed with compressor performing with high temperature differences between evaporating and condensing temperatures.

2.1. Comparative study of the performance of 3 stage compression system, multiple refrigerant injection system, and oil flooding system

This comparison was performed in terms of compressor COP, compressor work, and mixing effectiveness of compressor in case of OFS. The thermophysical properties of the refrigerant were calculated using EES software for all the three systems.

2.1.1. Thermodynamic Model for 3SCS

To calculate the system COP and compressor power consumption, the system requires to specify the evaporating and condensing temperatures, compressor isentropic efficiency, refrigerant, and refrigerant mass flow rate to get the thermophysical properties from EES. Analysis of flash tank was performed using mass and energy balance.

$$\dot{m}_{int} h_{10} + \dot{m}_R h_2 = \dot{m}_R h_{11} + \dot{m}_{int} h_3 \tag{1}$$

$$\dot{m}_c h_8 + \dot{m}_{int} h_4 = \dot{m}_{int} h_9 + \dot{m}_c h_5 \tag{2}$$

Where, $\dot{m}_{int}$ is the mass flowrate through the intermediate line, $\dot{m}_R$ is the mass flowrate through the evaporator and $\dot{m}_c$ is the total mass flowrate through the condenser. To calculate the intermediate pressure, equation is:

$$p_{1st\ int} = \sqrt{(p_{evap} * p_{2nd\ Int})} \tag{3a}$$

$$p_{2nd\ int} = \sqrt{(p_{1st\ Int} * p_{cond})} \tag{3b}$$

Where, p_{int} is the intermediate pressure, p_{evap} is the evaporator pressure, and p_{cond} is the condenser pressure. COP is the ratio of cooling capacity to compressor work. Compressor work and cooling capacity in case of 3SCS is as follows:

$$W_{comp} = \dot{m}_R(h_2 - h_1) + \dot{m}_{int}(h_4 - h_3) + \dot{m}_c(h_6 - h_5) \tag{4}$$

$$Q_{evap} = \dot{m}_R(h_1 - h_{12}) \tag{5}$$

$$COP = \frac{Q_{evap}}{W_{comp}} \tag{6}$$

Relative COP is the ratio of COP of cycle with multiple compressors to the COP of the cycle with single compressor.

$$COP_{Relative} = \frac{COP_{Multiple\ Comp}}{COP_{Simple}} \tag{7}$$

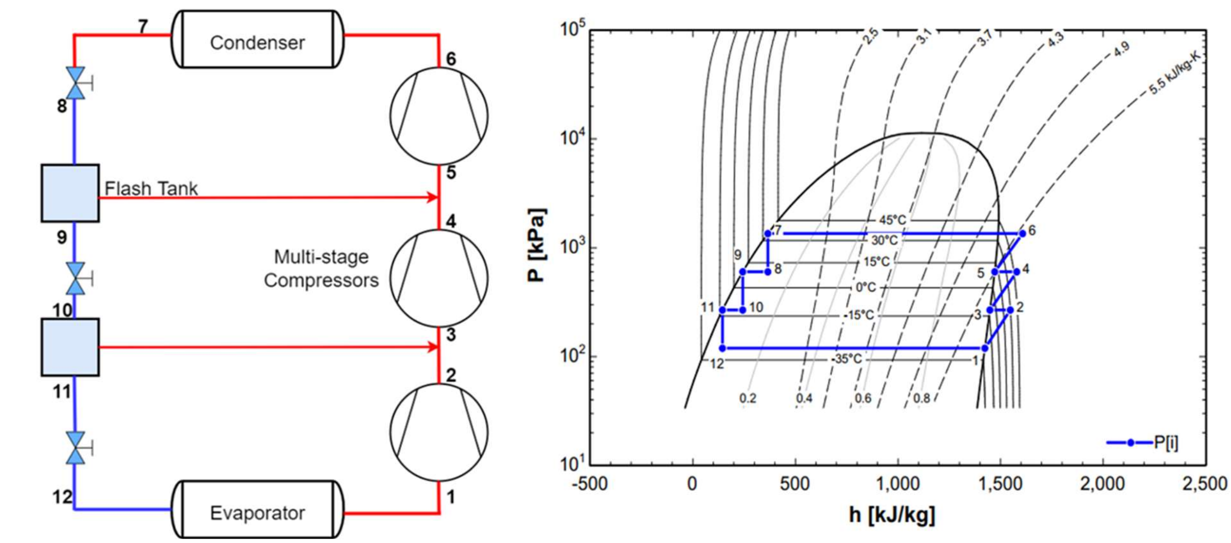

Fig. 1: 3 Stage Compression System and P-h Diagram for R410A

2.1.2. Thermodynamic Model for MRIS

The intermediate pressures of a multistage compressor are chosen to produce equal pressure ratios across the compressor stages, reducing compressor power. As a result, the injection pressures are chosen so that the pressure ratios between the injection points are equal. These injection pressures will be used same as intermediate pressure in the multi-stage compression system to estimate COP of the system with the economization.

To estimate the relative COP and COP of the MRIS, equation (6) and (7) were used, respectively.

The mass flow rate to the compressor at multiple injection points is defined as a fraction of the total mass flow rate. It was calculated using mass and energy balance.

$$\frac{\dot{m}[N_{inj}]}{\dot{m}_c} = \frac{h_6[N_{inj}]-h_3[N_{inj}]}{h_7[N_{inj}]-h_2[N_{inj}]} \tag{8}$$

$$\dot{m}_{inj}[N_{inj}] * h_9[N_{inj}] = \dot{m}_c * h_6[N_{inj}] - \dot{m}[N_{inj}] * h_7[N_{inj}] \tag{9}$$

$$\dot{m}_c = \dot{m}[N_{inj}] + \dot{m}_{inj}[N_{inj}] \tag{10}$$

Compressor work for MRIS was calculated as:

$$W_c[N_{inj}] = \dot{m} * \left(h_2[N_{inj}] - h_3[N_{inj} - 1]\right) + \dot{m}_c * (h_4[1] - h_3[N_{inj}]) \tag{11}$$

From isentropic efficiency, the enthalpies at States 2 and 4 are related to those at States 1 and 3.

$$\eta_{isen} = \frac{(h_{2s}[N_{inj}]-h_3[N_{inj}-1])}{(h_2[N_{inj}]-h_3[N_{inj}-1])} \tag{12}$$

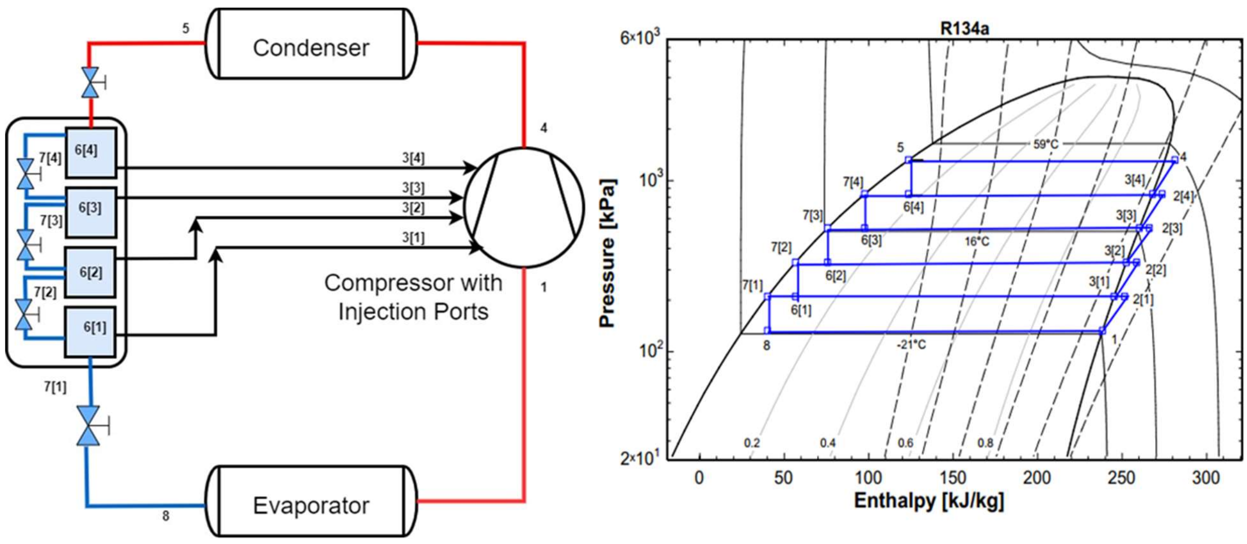

Fig. 2:MRIS with P-h Diagram for R410A [Mathison *et al*., (2010)]

2.1.2. Thermodynamic Model for OFS

Flooded compression cycle consists of the following components: Compressor, condenser, evaporator, Oil separation loop, expansion valve, regenerator. Flooded compression cycles are modelled similarly to normal, one-fluid vapor compression systems. The main changes are due to the oil and its qualities being considered throughout the compression process and in the oil loop. In order to evaluate the mixing qualities, certain simplifying assumptions are used. The phases are considered to exit the oil separator entirely separated, with no refrigerant dissolved in the oil stream or oil droplets carried in the refrigerant vapor stream.

The mixing model for the refrigerant and flooding agent, is the mixture enthalpy and mixture entropy as functions of pressure and temperature. The mixture properties can be expressed as a function of pressure, temperature, and flooding agent mass flow fraction, considering it the homogeneous mixture as:

$$h_m(p,T,X_L) = X_L h_L + (1 - X_L)h_G \tag{13}$$

$$S_m(p,T,X_L) = X_L S_L + (1 - X_L)S_G \tag{14}$$

Where, X_L is the flooding agent mass flow fraction, defined as the weighted average of the properties of the separated flooding agent and refrigerant gas phases calculated at the given pressure and temperature. The oil mass flow fraction is the ratio of the flooding agent mass flow rate and total mass flow rate (Refrigerant and flooding agent mass flow rate), as:

$$X_L = \frac{\dot{m_L}}{\dot{m_L}+\dot{m_G}} \tag{15}$$

$$X_G = \frac{\dot{m_G}}{\dot{m_L}+\dot{m_G}} \tag{16}$$

$$X_L + X_G = 1 \tag{17}$$

Where, $\dot{m_L}$ is the mass flow rate of the flooding agent and $\dot{m_G}$ is the mass flow rate of the gas phase refrigerant. The COP of the system was calculated using equation (6). Relative COP in case of OFS is the ratio of COP of flooded system to the COP of the non-flooded base system.

$$COP_{relative} = \frac{COP_{flooded}}{COP_{base}} \tag{18}$$

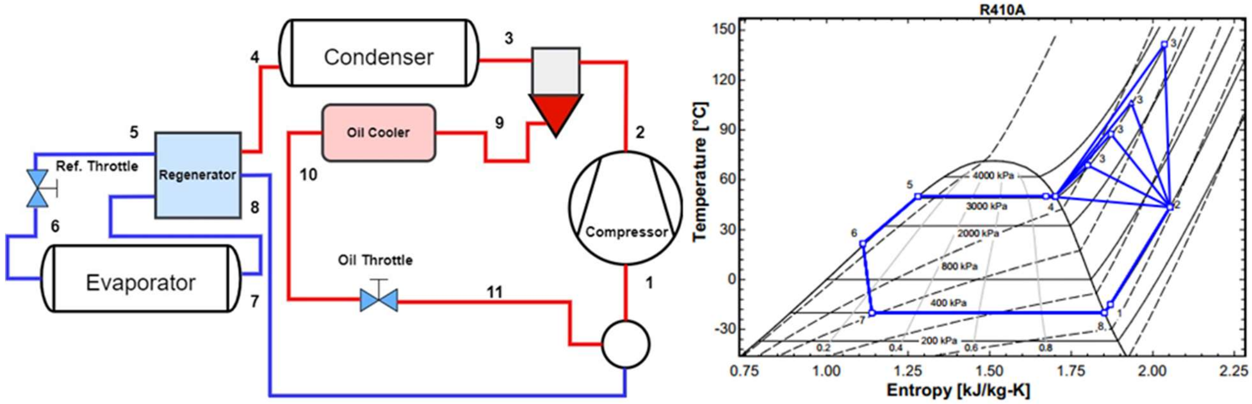

Fig. 3**:** OFS with T-s Diagram for R410a [Bell *et al*. (2011)]

3. RESULTS AND DISCUSSION

To compare the performance of all these 3 systems, the operating parameters given in table 1 was used. The ambient and condenser temperature were fixed at the same value for all these systems for fair comparison. Condensing temperature was kept at 50℃ in order to see the performance of all these 3 systems at high temperature difference between condensing and evaporating temperature. For this study only 2 refrigerants were kept under analysis: R410A and R454B. To reduce the environmental impact of R410A, R454B due to low GWP, has the lowest environmental impact was used as an alternative. However, the performance of R454B is lower than R410A in terms of energy efficiency and coefficient of performance (Panato *et al.* 2022). PAG oil properties were used as the flooding liquid for both refrigerants in OFS to ensure a reasonable comparison between R410A and R454B.

Table 1: Nominal values used in modeling

Parameter	Value
$\dot{m}_G$ [kg/s]	1
$T_{condenser}$ [℃]	50
η_{isen} [−]	0.7
T_o [℃]	25
$\varepsilon_{compressor}$ [-]	1

3.1 Comparison of Compressor Work and Compressor COP

Fig. 4 depicts the compressor work as a function of the evaporating temperature at constant condensing temperature for all systems: 3SCS, MRIS, and OFS. Comparing the compressor work, Fig. 4 shows that multistage compressors consume more work than MRIS and OFS. At evaporating temperature of -40℃, the multistage compressor consumes 54.54% more work than OFS, While OFS consumes 57.14% more work than MRIS.

In case of 3SCS, the system uses three compressors to operate in the same range of condensing and evaporating temperature, which sums up to absorb more energy, but cooling capacity is better than MRIS and OFS. As per quantified results, the cooling capacity for 3SCS, MRIS and OFS at the given operating conditions in table 1 for refrigerant R454B are 305KW, 240.4KW, and 160.1KW respectively. In case of MRIS and OFS, the heat of compression is mostly absorbed by injected fluid or oil due to which compression work reduces significantly as shown in Fig. 4.

Fig. 4 also presents the compressor performance for refrigerant R410A and R454B, and both refrigerants show the same trend in case of compressor power consumption. The sole difference noted here is that the power consumption in case of the R454B is more than R410A for all three systems, which is also observed from the study by Panato *et al.* (2022).

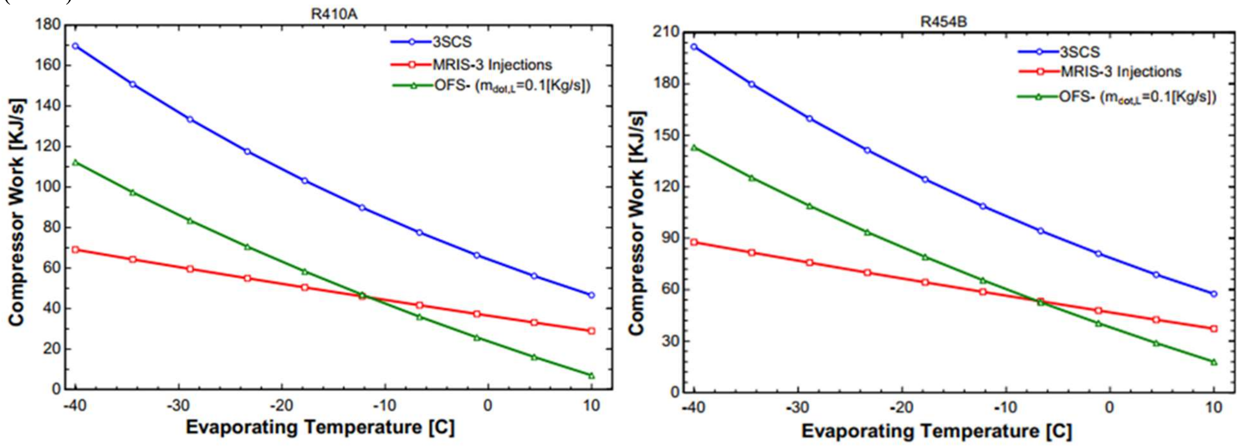

Fig. 4: Compressor work as a function of Evaporating temperature for R410A and R454B

The primary purpose of the current study is the performance comparison of the three systems in terms of COP of the system. Relative COP for MRIS is defined as the ratio of COP of the system with multiple refrigerant injections to COP of the baseline system.

$$COP_{relative} = \frac{COP_{MRI}}{COP_{baseline}} \tag{19}$$

The relative COP of the MRIS increases with the increase in the number of injection points. Fig. 5 presents the relative COP of the system as a function of evaporating temperature, which shows that relative COP is much higher for MRIS with 3 injections as compared to other two systems operating at the defined operating conditions mentioned in table 1. At evaporating temperature of -40°C, relative COP for MRIS with 3 injection points is 3.5% higher than 3SCS and 5.88% higher than OFS with refrigerant R410A. The same trend is followed by R454B operating under the same conditions mentioned in table 1.

Fig. 5 depicts that these three systems are applicable for high temperature lift applications because when the evaporating temperature increases at constant condensing temperature of 50°C, the temperature lift decreases, eventually reducing the relative COP for all the systems. Hence, the performance of these systems becomes less significant at low temperature lift. Figure 5 shows a range of evaporating temperatures that includes most of the operating conditions that a typical R-410A (R454B) application in domestic air conditioning equipment would see.

Comparing the performance of R454B with R410A, Fig. 5 shows that systems COP drops slightly with low GWP refrigerant R454B with the same operating conditions, which shows that this refrigerant can be used as a direct replacement of R410A with low environmental impact.

In case of OFS, working fluid selection and operating temperatures influence the impact of oil flooding on system efficiency. Because they are susceptible to severe compression and de-superheating losses, oil flooding is especially helpful for high heat of compression working fluids operating at large pressure ratios. The losses for the baseline cycle are primarily compression, throttling, and de-superheating losses at lower condensing temperatures. Flooding predominantly reduces the condenser's de-superheating losses, which is why the increase in cycle performance is greatest at the lowest source temperatures. Fig. 5 also shows that when evaporating temperature approaches 10°C, the COP of flooded compressor approaches the baseline system COP, which means, at this condition there is no significance of the flooded system.

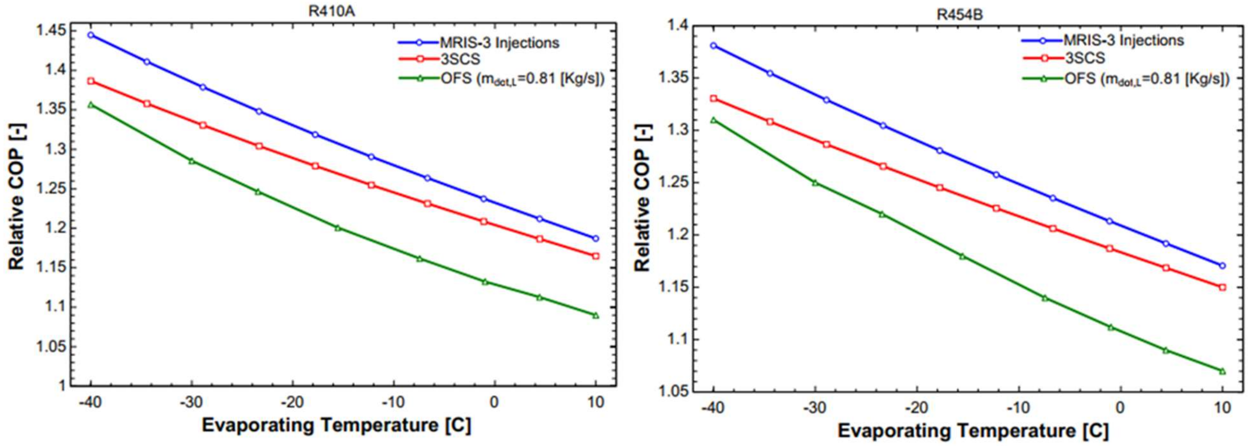

Fig. 5: Relative COP as a function of evaporating temperature for R410A and R454B

3.3 Mixing Effectiveness of Compressor for OFS

Effectiveness is defined as the ratio of actual heat transfer to the maximum heat transfer. Mixing effectiveness of compressor is the novelty of this work being introduced to the oil flooding system.

In all the results above, it is assumed that oil and refrigerant are in thermal and mechanical equilibrium, thus assuming that oil and refrigerant phases are both at the same temperature and pressure at the suction and discharge of the compressor, which means that the effectiveness of the compressor is equal to 1. In some cases, it may not be appropriate to assume perfect mixing and to evaluate how sensitive compressors are to non-ideal mixing, concept of mixing effectiveness has been introduced.

This study presents quantification of the influence of the non-ideal mixing model assumption on the resulting cycle performance by changing the refrigerant discharge temperature using the equation:

$$\varepsilon = \frac{T_{o,o} - T_{o,i}}{T_{r,o} - T_{o,i}} \tag{18}$$

Where, ε stands for effectiveness of compressor, $T_{o,o}$ stands for oil outlet temperature, $T_{o,i}$ stands for oil inlet temperature, and $T_{r,o}$ stands for the refrigerant outlet temperature.

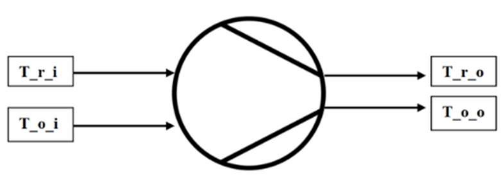

Fig. 6: Compressor Model

The effectiveness of the compressor is a function of refrigerant temperature at compressor discharge. In practical applications it is not possible to keep the suction or discharge thermal equilibrium, for oil and refrigerant phases and it is concluded that changing the refrigerant discharge temperature significantly drops the effectiveness of the compressor. For each 20°C increases in refrigerant temperature at compressor discharge drops the compressor effectiveness by 16% and so on. Fig. 6 shows the compressor model with the refrigerant and oil, suction and discharge conditions.

Fig. 7 depicts the compressor work and flooded system COP as a function of evaporating temperature. As can be seen from the plots, when the compressor effectiveness is equal to 1, the compressor work requirement is minimal and the system COP is maximum, whereas when the compressor effectiveness is less than or equal to 0.8, the compressor power consumption increases, and system COP decreases. For effectiveness equal to 0.9, the percent rise in compressor power consumption is 10.71 and likewise the reduction in system COP, while for effectiveness equal to 0.8, the compressor power consumption increases by 21%.

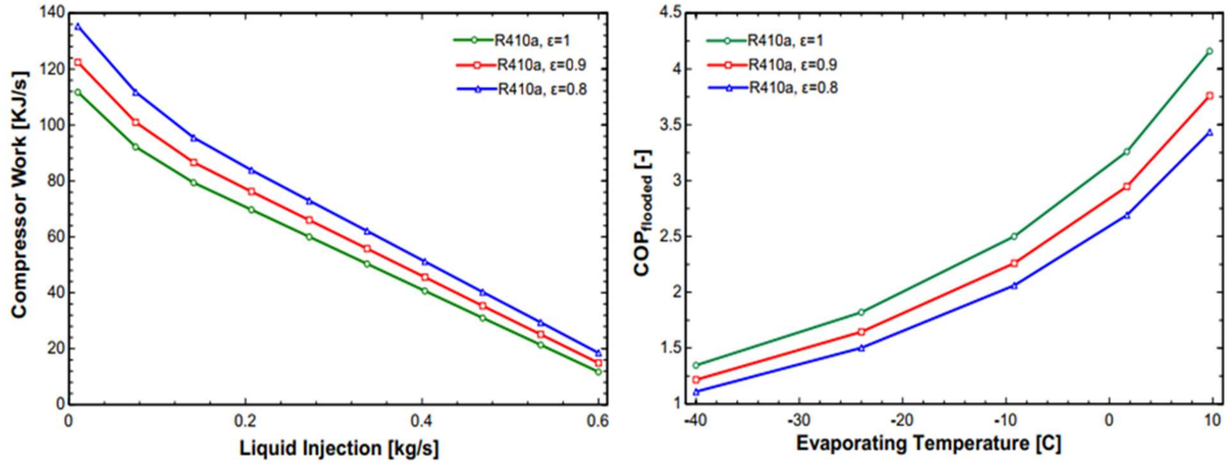

Fig. 7: Compressor work and Flooded COP as a function of evaporating temperature for R410A

Similarly, the compressor work and system COP trends are identical for refrigerant R454B and R410A as shown in Fig. 8. As previously stated, the system COP is slightly lower and compressor power consumption is higher for refrigerant R454B.

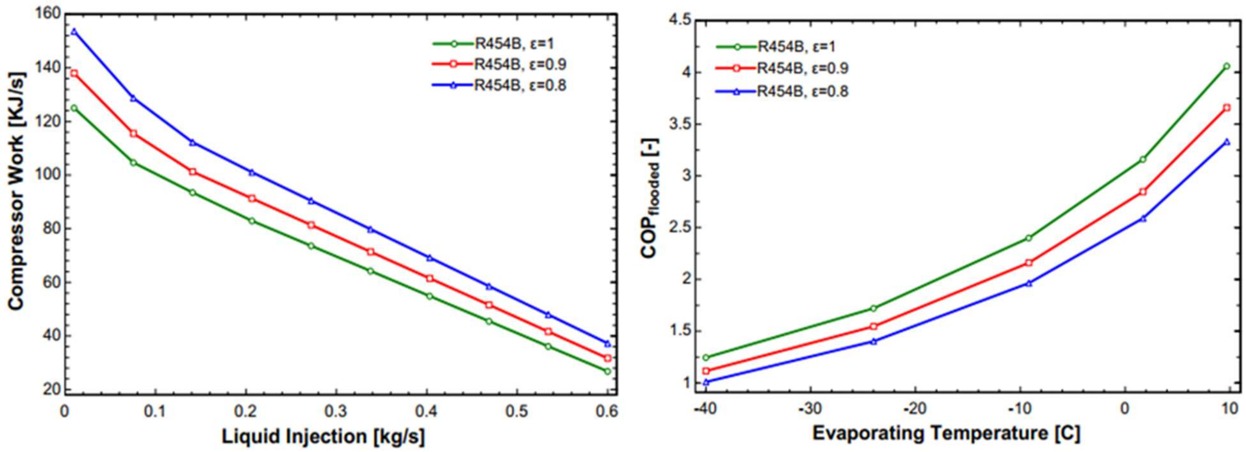

Fig. 8: Compressor work and Flooded COP as a function of evaporating temperature for R454B

4. CONCLUSIONS

The comparative analysis of these three systems: 3SCS, MRIS, and OFS were carried out at same operating conditions and results showed that all of them are appropriate for high temperature lift applications. The comparative performance demonstrated that keeping the condensing temperature constant and increasing the evaporating temperature reduces the temperature lift and likewise the significance of all these systems. The limit to the performance for each system is achieved and it has been observed that MRIS with 3 injection points outperform both 3SCS and OFS in terms of compressor work and system COP. Of the refrigerants considered, the performance of R410A and R454B observed to be same, with a tiny drop in COP for R454B.

The innovative concept of compressor mixing effectiveness in flooded compressors shown that when non-ideal mixing model assumptions are quantified by decreasing the compressor effectiveness from 1 to 0.8 for OFS, system

COP dropped by 21%, indicating a sensitivity that is proportional to the mixing efficiency of oil and refrigerant in an OFS system.

NOMENCLATURE

$\dot{m}_{int}$	Mass flow rate through the intermediate line	[kg/s]
$\dot{m}_{R}$	Mass flow rate through the evaporator	[kg/s]
$\dot{m}_{c}$	Mass flow rate through the condenser	[kg/s]
p_{int}	Intermediate Pressure	[KPa]
$T_{o,o}$	Oil outlet temperature	[°C]
$T_{o,i}$	Oil inlet temperature	[°C]
$T_{r,o}$	Refrigerant outlet temperature	[°C]
X_G	Flooding agent mass flow fraction	[-]
MRIS	Multiple refrigerant injections system	
3SCS	Three stage compression system	
OFS	Oil flooding system	
COP	Coefficient of performance	
GWP	Global warming potential	

Greek Symbols

η_{isen}	Compressor isentropic efficiency
Ɛ	Compressor effectiveness

Subscripts

m	mixture
G	Gas refrigerant
L	Liquid oil

REFERENCES

Agrawal, N., & Bhattacharyya, S. (2007). Studies on a two-stage transcritical carbon dioxide heat pump cycle with flash intercooling. *Applied Thermal Engineering*, *27*(2–3), 299–305. https://doi.org/10.1016/j.applthermaleng.2006.08.008

Bell, I. H., Groll, E. A., & Braun, J. E. (2011). Performance of vapor compression systems with compressor oil flooding and regeneration. *International Journal of Refrigeration*, *34*(1), 225–233. https://doi.org/10.1016/j.ijrefrig.2010.09.004

Bertsch, S. S., & Groll, E. A. (2008). Two-stage air-source heat pump for residential heating and cooling applications in northern U.S. climates. *International Journal of Refrigeration*, *31*(7), 1282–1292. https://doi.org/10.1016/J.IJREFRIG.2008.01.006

Cabello, R., Torrella, E., Llopis, R., & Sánchez, D. (2010). Comparative evaluation of the intermediate systems employed in two-stage refrigeration cycles driven by compound compressors. *Energy*, *35*(3), 1274–1280. https://doi.org/10.1016/j.energy.2009.11.009

Coney, M. W., Stephenson, P., Plc, I., Linnemann, C., & Morgan, R. E. (2002). *Development Of A Reciprocating Compressor Using Water Injection To Achieve Quasi-Isothermal Compression A. Malmgren*. Retrieved from https://docs.lib.purdue.edu/icec

Dalkiliç, A. S. (2011). *AJTEC2011-44267 COMPARISON OF VARIOUS ALTERNATIVE REFRIGERANTS FOR VAPOUR COMPRESSION REFRIGERATION SYSTEMS*. Retrieved from http://www.asme.org/about-asme/terms-of-use

Feng, C., Kai, W., Shouguo, W., Ziwen, X., & Pengcheng, S. (2009). Investigation of the heat pump water heater using economizer vapor injection system and mixture of R22/R600a. *International Journal of Refrigeration*, *32*(3), 509–514. https://doi.org/10.1016/J.IJREFRIG.2008.06.012

Guoyuan, M., Qinhu, C., & Yi, J. (2003). Experimental investigation of air-source heat pump for cold regions. *International Journal of Refrigeration*, *26*(1), 12–18. https://doi.org/10.1016/S0140-7007(02)00083-X

Hugenroth, J., Braun, J., Groll, E., King, G., & Braun, J. (n.d.). *Purdue e-Pubs Liquid-Flooded Ericsson Cycle Cooler: Part 1-Thermodynamic Analysis LIQUID-FLOODED ERICSSON CYCLE COOLER: PART 1-*

THERMODYNAMIC ANALYSIS. Retrieved from http://docs.lib.purdue.edu/iracc/823

Jiang, S., Wang, S., Jin, X., & Zhang, T. (2015). A general model for two-stage vapor compression heat pump systems. *International Journal of Refrigeration*, *51*, 88–102. https://doi.org/10.1016/J.IJREFRIG.2014.12.005

Ma, G. Y., & Zhao, H. X. (2008). Experimental study of a heat pump system with flash-tank coupled with scroll compressor. *Energy and Buildings*, *40*(5), 697–701. https://doi.org/10.1016/J.ENBUILD.2007.05.003

Mathison, M. M., Braun, J. E., Groll, E. A., Mathison, M. M. ;, & Braun, J. E. ; (n.d.). *International Refrigeration and Air Conditioning Conference at Purdue*. Retrieved from http://docs.lib.purdue.edu/iracc/1034

Panato, V. H., Marcucci Pico, D. F., & Bandarra Filho, E. P. (2022). Experimental evaluation of R32, R452B and R454B as alternative refrigerants for R410A in a refrigeration system. *International Journal of Refrigeration*, *135*, 221–230. https://doi.org/10.1016/j.ijrefrig.2021.12.003

Patil, V. C., Acharya, P., & Ro, P. I. (2020). Experimental investigation of water spray injection in liquid piston for near-isothermal compression. *Applied Energy*, *259*. https://doi.org/10.1016/j.apenergy.2019.114182

Röyttä, P., Turunen-Saaresti, T., & Honkatukia, J. (2009). Optimising the refrigeration cycle with a two-stage centrifugal compressor and a flash intercooler. *International Journal of Refrigeration*, *32*(6), 1366–1375. https://doi.org/10.1016/J.IJREFRIG.2009.01.006

Wang, X., Hwang, Y., & Radermacher, R. (2009). Two-stage heat pump system with vapor-injected scroll compressor using R410A as a refrigerant. *International Journal of Refrigeration*, *32*(6), 1442–1451. https://doi.org/10.1016/J.IJREFRIG.2009.03.004

Xu, S. Z., Wang, L. W., & Wang, R. Z. (2016). Thermodynamic analysis of single-stage and multi-stage adsorption refrigeration cycles with activated carbon-ammonia working pair. *Energy Conversion and Management*, *117*, 31–42. https://doi.org/10.1016/j.enconman.2016.03.010

Xu, X., Hwang, Y., & Radermacher, R. (2011). Refrigerant injection for heat pumping/air conditioning systems: Literature review and challenges discussions. *International Journal of Refrigeration*, *34*(2), 402–415. https://doi.org/10.1016/J.IJREFRIG.2010.09.015

ACKNOWLEDGEMENT

This work has been supported by the Center for Integrated Buildings Systems (CIBS) at Oklahoma State University.