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## Performance Limit for Economized Cycles with Continuous Refrigerant Injection

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### ABSTRACT

The cost of multi-stage compressors has traditionally limited the implementation of economization to larger-scale applications. However, the development of compressors with ports for refrigerant injection, which can be added at relatively low cost, introduces new opportunities for economizing in small-scale applications. Injecting economized refrigerant provides cooling during the compression process, and thus reduces the work required for compression and improves the cycle's coefficient of performance (COP) despite a slight decrease in cooling capacity. A simple model is developed to show that increasing the number of injection ports improves COP, approaching a performance limit defined by continuous injection. In the case of continuous injection, refrigerant in the compressor remains saturated vapor throughout the compression process due to the cooling provided by the injected refrigerant. The model predicts that continuous injection will improve the COP between 18% and 51% for common air-conditioning and refrigeration applications, benefiting higher temperature lift cycles most significantly.

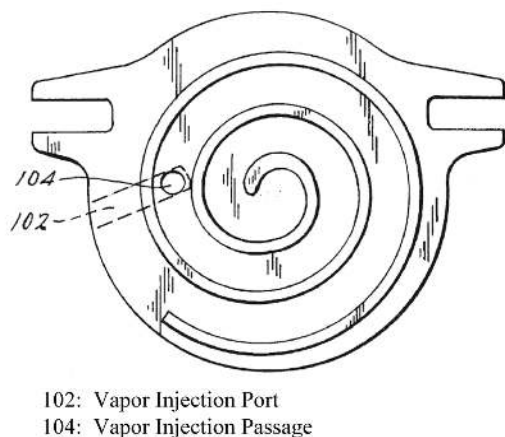
### 1. INTRODUCTION

#### 1.1 Background

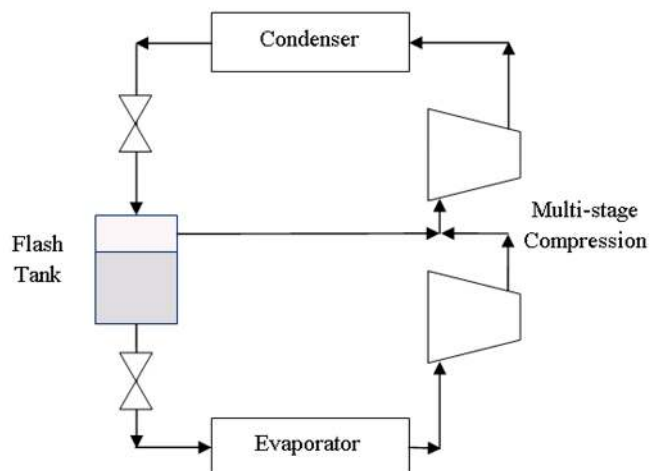
Because of the widespread use of vapor compression cycles in heating and cooling applications, a small improvement in the efficiency of vapor compression equipment can have a significant impact on total energy use. While the development and use of high performance components, such as compressors and expanders, can improve the cycle performance, the cost and complexity of these components can preclude their implementation in the marketplace. Modifications to the basic cycle configuration offer an alternative approach for providing performance improvements. Common cycle modifications include the incorporation of intercooling and economizing, which both aim to reduce the compressor power consumption and, in some cases, improve cooling capacity.

Traditionally, the incorporation of intercooling or economizing has required the use of a multi-stage compressor, the cost of which has limited the scope of these modified cycles to large-scale applications. However, the development of compressors with ports for refrigerant injection during the compression process has introduced new possibilities for cycle modification. Figure 1 shows a scroll compressor with a single refrigerant injection port patented by Copeland Corporation (Perevozchikov, 2003). While the injection port cannot be used to provide intercooling, economized refrigerant can be injected through the port to essentially break the compression process into two stages within a single scroll. Refrigerant injection could similarly be implemented within a rolling piston compressor, providing a relatively simple, low cost alternative to multi-stage compressors.

The goal of economizing is 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. This decreases the specific work required for compression. Alternative cycle modifications have been proposed to achieve this compressor cooling through different means. For example, oil flooding of the compressor



**Figure 1. Scroll compression with refrigerant injection (Perevozchikov, 2003).**



**Figure 2. Vapor compression cycle with two-stage compression and flash-tank economization.**

improves cycle performance by using a high heat capacity fluid to absorb the heat of compression and approach an isothermal compression process (Bell *et al.*, 2010). However, this introduces complexity related to oil separation and oil cooling. In contrast, economization uses the injected refrigerant to absorb the heat of compression. Previous researchers have demonstrated the potential to reduce power consumption through economization with multi-stage compression both experimentally and analytically (Bertsch and Groll, 2008; Cho *et al.*, 2009; Torrella *et al.*, 2009; Wang *et al.*, 2009; Winandy and Lebrun, 2002). One common configuration for achieving economization with a two-stage compressor is shown schematically in Figure 2.

In the configuration depicted in Figure 2, to be referenced as flash-tank economization, the refrigerant exiting the condenser expands to an intermediate pressure and enters a phase separator. The phase separator can theoretically supply either a saturated vapor or a saturated liquid-vapor mixture to the mixing point between the compressor stages. In both cases it is assumed that the injection line draws off all vapor generated by the expansion process, leaving saturated liquid refrigerant at the entrance to the expansion valve that leads to the evaporator. An alternative configuration uses an internal heat exchanger (IHX) in place of the phase separator. While the use of a heat exchanger with non-mixing streams may provide greater control over the system, the heat transfer in the IHX cannot match the effectiveness of heat transfer in a flash tank. Therefore, this study focuses on the higher performing configuration of flash-tank economization.

Previous research studies have demonstrated the effectiveness of economization, whether using a flash-tank or IHX configuration. Winandy and Lebrun (2002) and Torrella *et al.* (2009) both calculated an increase in COP with IHX economization based on experimental results. Cho *et al.* (2009) performed similar testing on a transcritical carbon dioxide cycle, reporting a 16.5% increase in the cycle COP compared to the cycle without vapor injection. Wang *et al.* (2009) experimentally investigated the performance of a vapor-injected scroll compressor with both flash tank and IHX economization, and observed similar performance improvements for both configurations in heating and cooling modes. Other studies have focused on the performance of the vapor compression cycle in the heating mode, including an experimental and theoretical evaluation of a low-temperature heat pump with economization by Bertsch and Groll (2008).

All of the previously mentioned studies considered cycles that use two-stage compression or a compressor with one injection point. Jung *et al.* (1999) extended this concept and used an experimentally validated model to show that a cycle with three-stage compression and two economizers has a higher COP than a cycle with two-stage compression and one economizer. Therefore, introducing additional stages or additional injection points has the potential to provide even greater improvements in cycle COP and capacity.

Of course, introducing additional compressor staging increases the cost and complexity of both the compressor and the overall cycle. This not only limits the number of compressor stages that can be practically implemented, but also prohibits the incorporation of multi-stage compressors in small-scale applications, as mentioned previously.

However, the development of compressors with refrigerant injection ports introduces the opportunity to essentially create multiple stages within a single compression chamber. Instead of injecting the economized refrigerant between stages, the economized refrigerant is injected during the compression process, providing the cooling benefits of economizing without the cost of a multi-stage compressor.

In addition, the number of injection points can be increased relatively easily and at little additional cost, whereas the incorporation of additional compressor stages carries a significant penalty of cost and complexity. Each additional injection point provides more cooling to the compressed refrigerant, maintaining the refrigerant closer to the saturated vapor state throughout the compression process and thus reducing the compression work. Therefore, the performance of the vapor compression cycle with economizing should increase with the number of injection ports, just as Jung *et al.* (1999) demonstrated an increase in cycle COP with increased compressor staging.

## 1.2 Motivation

The potential to improve the performance of vapor compression cycles through the injection of economized refrigerant during the compression process motivates this study. Figure 3 provides a schematic of a system with a compressor that has three refrigerant injection points; the economized refrigerant is supplied to the compressor by a phase separator device, selected over the IHX configuration for its effective heat transfer. Letting  $N$  represent the number of the injection points in the compressor, there will be a corresponding number of  $N$  injection lines and phase separators, and  $N+1$  expansion valves. The refrigerant states exiting the expansion valves and entering phase separators are identified as 6[1..N], where higher index numbers correspond to higher injection pressures. Similar indexing is used for the liquid streams exiting the phase separators, identified as States 7[1..N], and the two-phase injection streams exiting the phase separators, identified as States 9[1..N]. Because each injection line requires a phase separator, the complexity of the overall system increases significantly with each additional injection point despite the relative ease of incorporating injection ports in the compressor. Therefore, this study aims to provide a cycle model that can be used to select the number of injection ports that balance the tradeoff between improved cycle performance and system complexity.

A second goal of this study is to define an upper limit to economized cycle performance, which sets a goal for economization. The unmodified vapor compression cycle with single-stage compression provides a lower limit to the cycle performance, and the performance improvements due to economizing can be measured relative to this baseline case. The upper limit to economized cycle performance is defined by continuous injection, in which the injected refrigerant provides enough cooling to maintain a saturated vapor in the compression chamber. Therefore, the performance of the compressor with a finite number of injection points can also be judged by its ability to approach this limit on performance. In addition, determining the limit to maximum performance with economizing provides a tool for identifying the applications that show the most potential for improved performance through economizing.

## 2. ANALYSIS

The vapor compression cycle was modeled both with and without economizing modifications using Engineering Equation Solver (EES; Klein, 2009). The basic cycle model requires the user to specify the evaporating and condensing temperatures, degree of superheat at the compressor inlet, compressor isentropic efficiency, and degree of subcooling at the condenser outlet. Assuming that the throttling device operates adiabatically and neglecting

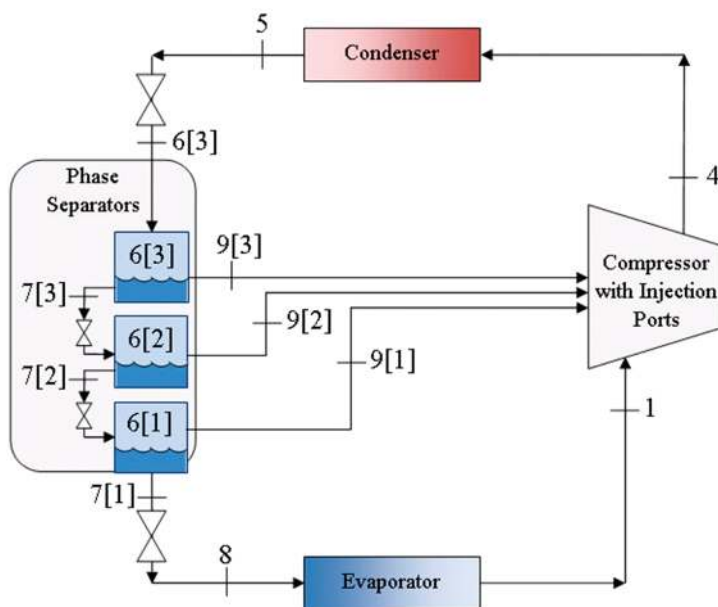
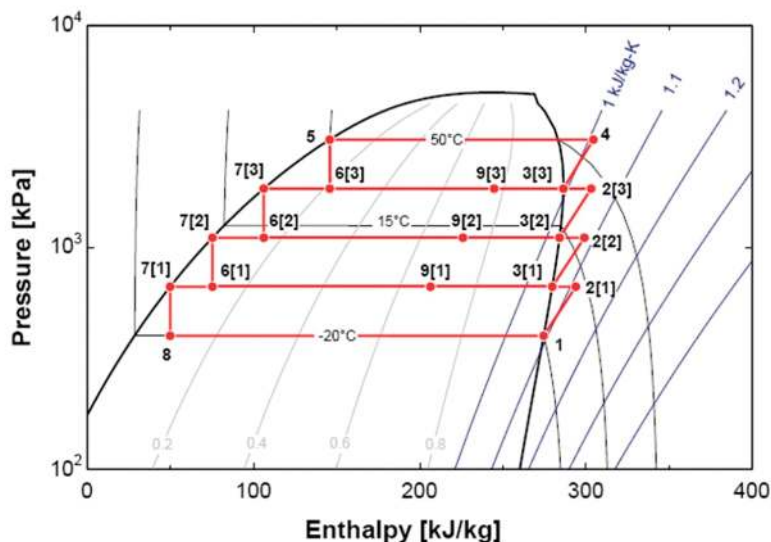


Figure 3. Vapor compression cycle with injection and flash-tank economization.

pressure drops in the heat exchangers, these parameters provide sufficient information for calculating the compressor power consumption, the cycle's heating or cooling capacity, and the COP of the cycle. The results of this study focus on the performance of the cycle in cooling mode due to the greater prevalence of vapor compression equipment in cooling applications.

The economizing configuration under consideration uses phase separators to supply a liquid-vapor mixture to the injection lines, as shown in Figure 4. It is assumed that the phase separators can be controlled such that a two-phase mixture with a specified quality is supplied to the injection line, while saturated liquid is supplied to the downstream expansion valve. To achieve this control in practice, it may be necessary to first separate the liquid and vapor phases, and then mix the desired amount of liquid into the vapor stream; however, the specific design of this component is beyond the scope of the current project, which aims to evaluate only the potential benefits of the cycle modifications. The desired quality of the injected refrigerant is determined by assuming that the injected refrigerant mixes instantaneously with the refrigerant in the compressor and results in a saturated vapor state. However, the pressure in each injection line must be specified to solve the mass and energy balances on the mixing process.



**Figure 4. Pressure-enthalpy diagram for a vapor compression cycle with three ports for injecting economized refrigerant in the compressor.**

## 2.1 Model Development: Finite Number of Injection Points

In a multi-stage compressor, the intermediate pressures are selected to result in equal pressure ratios across the compressor stages, which minimize the compressor power (Moran and Shapiro, 2000). Therefore, the injection pressures are also selected to result in equal pressure ratios between the injection points. An initial analysis showed that this does not correspond precisely to the point of minimum work due to the change in mass flow through the compressor, but is sufficient for the current model. It will result in the model slightly underpredicting the improvements in COP with economization when there are a small number of injection points, but will become insignificant as the number of injection points is increased to approach the limiting case of continuous injection.

Figure 4 shows the states of the refrigerant in an economized cycle with three injection points as calculated by the EES model. The labeled states correspond to those identified on the cycle diagram in Figure 3. The specified evaporating temperature of  $-20^{\circ}\text{C}$  and the evaporator superheat of  $0^{\circ}\text{C}$  fix the refrigerant properties at State 1, entering the compressor. The injection pressures are calculated by the method discussed previously, using the specified condensing temperature of  $50^{\circ}\text{C}$  to determine the condensing pressure. Once the injection pressures are known, the isentropic efficiency of the compressor, specified as 70%, fixes the properties at the end of each compression process (States 2[1..N] and State 4), where  $N$  represents the number of injection points. States 3[1..N] are the saturated vapor states resulting from the instantaneous mixing processes at the respective injection pressures.

The subcooling at the condenser exit, or in this case the specification of a saturated liquid, fixes the properties of the refrigerant at the condenser exit, State 5. The refrigerant then expands in an isenthalpic process to the phase separator operating at the highest injection pressure, State 6[3], where the enthalpy and pressure fix the state properties. State 9[3] represents the refrigerant leaving this phase separator through the injection line. Because it is assumed that all of the vapor generated during the expansion process is directed to the injection line, only saturated liquid remains in the main refrigerant line, State 7[3], leading towards the next expansion process. A mass balance on the separation process relates the mass flow rate in the main refrigerant line at State 7[3] to the mass flow rate in the injection line at State 9[3]. The mass flow rates through these lines are defined as a fraction of the total mass

flow rate through the condenser, where  $r_{comp}[i]$  is the fraction of the total mass flow rate that exits the compressor at pressure  $p_{inj}[i]$ , and  $r_{inj}[i]$  is the fraction of the total mass flow rate through the injection line at pressure  $p_{inj}[i]$ :

$$r_{comp}[i] = \frac{\dot{m}_{comp}[i]}{\dot{m}_{cond}} \quad (1)$$

$$r_{inj}[i] = \frac{\dot{m}_{inj}[i]}{\dot{m}_{cond}} \quad (2)$$

While both of the mass fractions are unknown, combining the mass balance with the energy balances on the separation and mixing processes provides sufficient information to solve for the mass fractions as well as the enthalpy,  $h$ , of the injected refrigerant:

$$r_{comp}[i] = r_{comp}[i+1] \left( \frac{h_6[i] - h_3[i]}{h_7[i] - h_2[i]} \right) \quad (3)$$

$$r_{inj}[i] = r_{comp}[i+1] - r_{comp}[i] \quad (4)$$

$$h_9[i] = \frac{r_{comp}[i+1] \cdot h_6[i] - r_{comp}[i] \cdot h_7[i]}{r_{inj}[i]} \quad (5)$$

This calculation procedure can then be repeated for the expansion valve and phase separator combination at successively lower injection pressures. Finally, an isenthalpic expansion process from the lowest injection pressure to the evaporating pressure fixes the properties at the evaporator inlet, State 8.

## 2.2 Model Development: Approaching an Infinite Number of Injection Points

The model predicts the performance of the cycle under the limiting case of continuous injection by considering a large enough number of injection points that introducing additional points does not change the model results. Therefore, it is desirable to simplify the model where possible, beginning with writing Equations (3)-(5) in terms of the saturated liquid and saturated vapor properties of the refrigerant. The enthalpies at States 5, 6, 7, and 8 depend only on saturated liquid properties, whereas the enthalpies at States 1 and 3 depend on saturated vapor properties. The enthalpies at States 2 and 4 are related to States 1 and 3 by the definition of isentropic efficiency,  $\eta_s$ :

$$h_2[i] = h_3[i-1] + \frac{h_{2s}[i] - h_3[i-1]}{\eta_s} \quad (6)$$

In addition, the  $Tds$  equations relate the change in enthalpy,  $dh$ , to the change in entropy,  $ds$ , the change in pressure,  $dp$ , the temperature,  $T$ , and the specific volume,  $v$ , at any point (Moran and Shapiro, 2000):

$$Tds = dh - vdp \quad (7)$$

When the model is solving for the cycle performance in the limiting case or with a large number of injection points, the pressure step in the compressor is small and it can be assumed that the specific volume of the refrigerant remains nearly constant during the isentropic compression process from State 3[i-1] to State 2s[i]. Therefore, Equation (7) can be integrated to solve for the change in enthalpy during the isentropic compression process:

$$h_{2s}[i] - h_3[i-1] = v_3[i-1](p_{inj}[i] - p_{inj}[i-1]) \quad (8)$$

Substituting Equation (8) into Equation (6) provides an expression for the enthalpy at State 2 in terms of the properties at State 3:

$$h_2[i] = h_3[i-1] + \frac{v_3[i-1](p_{inj}[i] - p_{inj}[i-1])}{\eta_s} \quad (9)$$

Because State 3 is the saturated vapor state, this expression provides a method for approximating the enthalpy at State 2 using saturated vapor properties. Therefore, Equation (9) can now be substituted back into Equation (3) to obtain an expression for the mass flow ratio that only depends on saturated liquid and saturated vapor properties, denoted by the subscripts  $f$  and  $g$ , respectively:

$$r_{comp}[i] = r_{comp}[i+1] \left\{ \frac{h_f[i+1] - h_g[i]}{h_f[i] - \left[ h_g[i-1] + \frac{v_g[i-1](p_{inj}[i] - p_{inj}[i-1])}{\eta_s} \right]} \right\} \quad (10)$$

While Equation (10) only applies to cycles with small steps in injection pressure, it provides a useful tool for building a model that only requires saturated liquid and saturated vapor properties. Such a model could be built in any programming environment with basic equations of state for saturated liquid and vapor properties, whereas Equation (3) requires superheated vapor properties that are more easily accessed in a program such as EES.

Equation (10) proves most useful for analyzing the limiting economization case with continuous injection. In this limiting case, the pressure step between each injection port becomes infinitesimally small, enabling the assumption of constant specific volume during each compression step. Equation (10) also shows the impact of the compressor isentropic efficiency on the limit to cycle performance; a less efficient compressor will require a larger mass flow rate of injected refrigerant to maintain the saturated vapor state in the compressor. Figure 5 plots the limiting case on a pressure-enthalpy diagram assuming that the compressor has an isentropic efficiency of 70%, saturated vapor enters the compressor and saturated liquid exits the condenser. The state of the refrigerant in the compressor follows the vapor curve, while the state of the refrigerant entering the expansion valves follows the liquid curve, as expected.

### 2.3 Model Results

Figure 6 presents the model results for an R-410A system with an increasing number of injection points; the cycle operates with the same superheat, subcooling, and compressor efficiency specified for Figure 5. Because the effect of economization is most pronounced for applications with a large temperature lift, the results in Figure 6 consider an evaporating temperature of  $-20^{\circ}\text{C}$  and condensing temperature of  $50^{\circ}\text{C}$ . The figure shows the improvement in cooling COP relative to the performance of the unmodified, single-stage vapor compression cycle.

The relative COP of the cycle increases with the incorporation of additional injection points as expected, approaching a maximum value for the COP as the number of injection points increases. Incorporating one injection point increases the COP 21% over its baseline value, while the second and third injection points improve the COP an additional 8% and 4% over the baseline value, respectively. The horizontal line in Figure 6 represents the upper limit to COP improvements, approximately 44%, that is possible with continuous refrigerant injection. The tradeoff between the cost and complexity incurred by the addition of each injection line and its contribution to the system's performance will determine how closely the actual system approaches this optimal design.

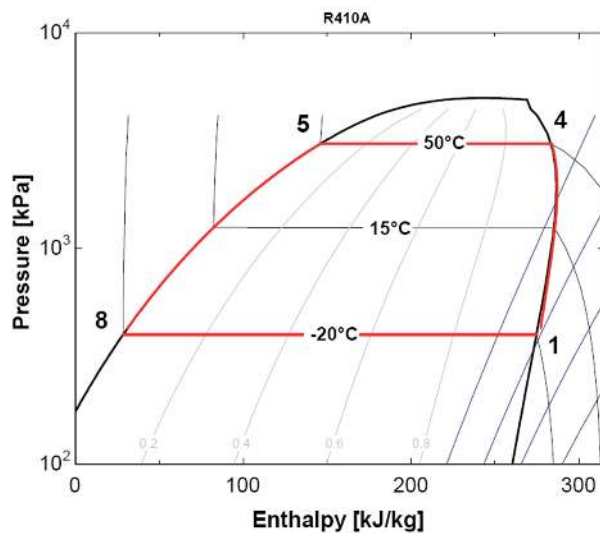


Figure 5. Pressure-enthalpy diagrams for a vapor compression cycle with continuous injection of economized refrigerant in the compressor.

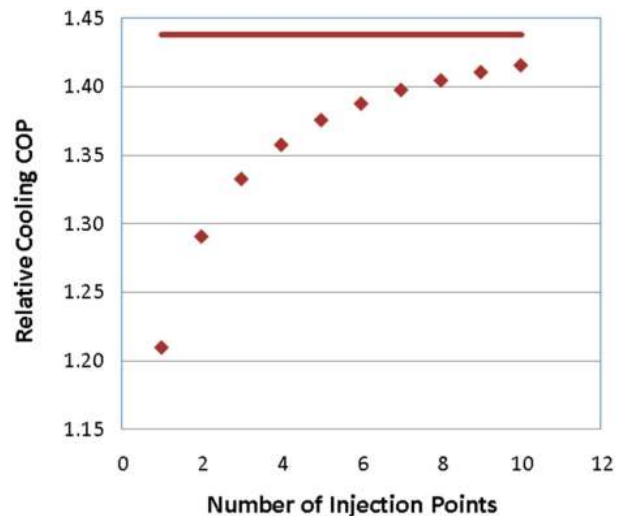


Figure 6. Variation in relative COP with number of injection points for R-410A cycle operating at evaporating temperature of  $-20^{\circ}\text{C}$  and condensing temperature of  $50^{\circ}\text{C}$ .

Of primary interest in this study is the performance limit of the economized cycle with continuous injection. For the conditions discussed previously, an R-410A cycle with an evaporating temperature of  $-20^{\circ}\text{C}$  and a condensing temperature of  $50^{\circ}\text{C}$ , the maximum improvement in COP was approximately 44%. However, Figure 7 shows that this performance improvement becomes less significant as the temperature lift decreases, whether due to an increase in evaporating temperature or a decrease in condensing temperature.

The range of evaporating and condensing temperatures plotted in Figure 7 covers most of the operating conditions that would be encountered by a typical application of R-410A in residential air conditioning equipment. However, refrigerators and freezers or automobile air-conditioning applications, which can encounter lower evaporating and condensing temperatures, typically use R-134a and therefore will respond differently to economizing.

Figure 8 plots the maximum improvement in COP with economization for an R-134a system with the same operating parameters. While the graph shows larger improvements in COP than for the R-410A system, this primarily results from the lower range of evaporating temperatures considered for R-134a systems. In fact, the graph shows that the improvement in COP drops significantly as the condensing temperature decreases from  $50^{\circ}\text{C}$  to  $40^{\circ}\text{C}$ , so the lower condensing temperature encountered in R-134a applications will likely make economizing less effective. Nonetheless, any system operating across a large temperature lift can benefit significantly from economizing.

Figure 9 presents results for the same cycle using R-404A, which is commonly found in supermarket refrigeration. The evaporating temperatures in these applications can be very low depending on the exact use of the display case (refrigerated versus frozen foods), and the condensing temperatures can be very high if the condensing unit operates outside. Therefore, the large temperature lift across the cycle makes supermarket refrigeration a prime target for economization. Figure 10 shows that R-404A performs very well with economization, and the COP is almost doubled in the extreme case of a  $-40^{\circ}\text{C}$  evaporating temperature and  $50^{\circ}\text{C}$  condensing temperature. However, part of the reason the R-404A cycle experiences such large

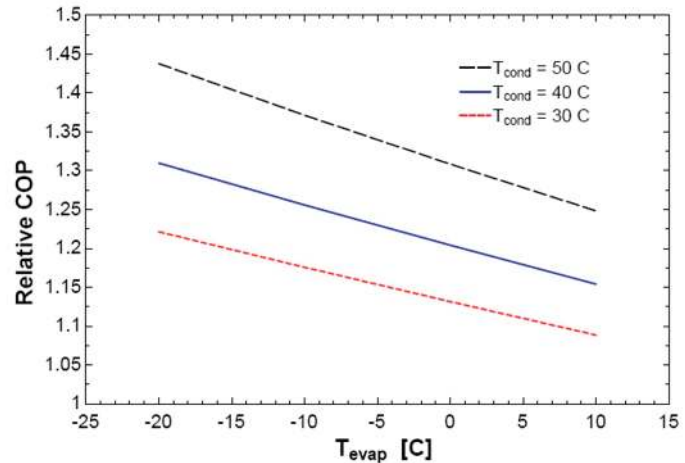


Figure 7. Maximum improvement in COP through economizing for an R-410A cycle.

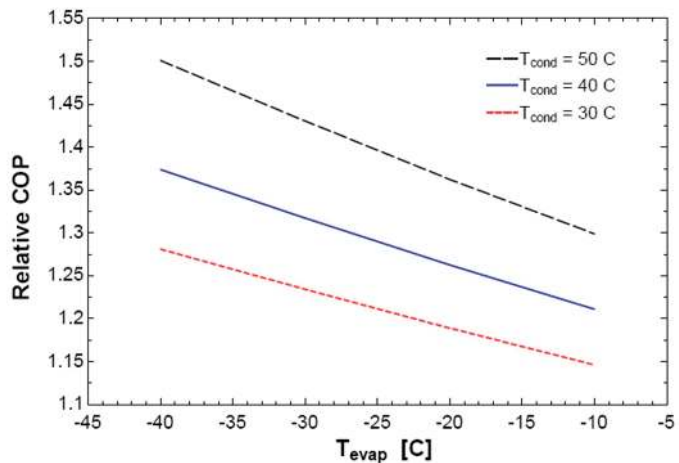


Figure 8. Maximum improvement in COP through economizing for an R-134a cycle.

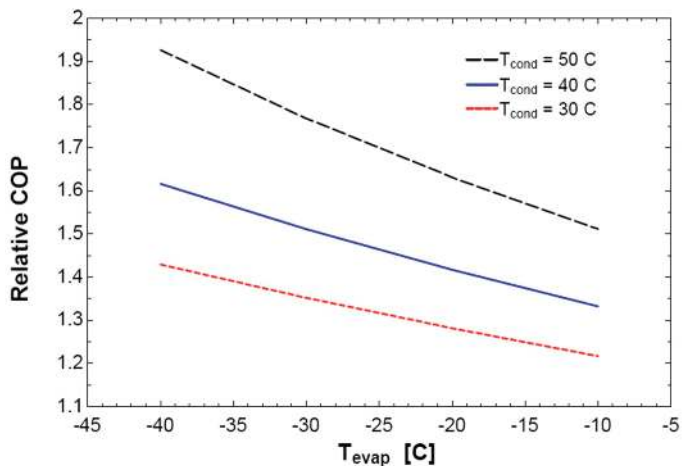


Figure 9. Maximum improvement in COP through economizing for an R-404A cycle.



improvements in COP with economization is the low COP of the basic cycle without economization. As a result of the low COP, many other cycle modifications have been proposed for supermarket applications, and the benefits of vapor injection must be considered relative to these other technologies.

### 3. CONCLUSIONS AND RECOMMENDATIONS

The development of compressors with refrigerant injection ports introduces the possibility of extending economizing to smaller scale applications. In addition, the benefits of economizing can be increased by introducing multiple injection ports in these compressors. The limit to performance improvements with economization is achieved when the continuous injection of economized refrigerant maintains a saturated vapor state in the compressor.

A basic EES model demonstrated that the maximum improvement in COP for this limiting case is approximately 18% for an R-410A cycle with an evaporating temperature of 5°C and a condensing temperature of 40°C. Similar performance improvements were predicted for an R-134a cycle; with an evaporating temperature of -20°C and a condensing temperature of 30°C, the model predicts a 22% improvement in COP for an R-134a cycle. However, the benefits of economizing become more pronounced with larger temperature lifts, which are common in supermarket refrigeration cycles using R-404A. Continuous injection can increase the COP of an R-404A cycle by 51% with an evaporating temperature of -30°C and a condensing temperature of 40°C. Therefore, R-404A is a particularly promising refrigerant for this technology. About 75% of the maximum benefit associated with continuous injection can be achieved with three injection points. Implementing a compressor with continuous refrigerant injection would require a much clearer understanding of the physical process in the compressor in order to achieve good refrigerant mixing and avoid damage due to liquid slugging. In addition, implementation of these compressors will require development of affordable phase separator devices that can be easily controlled to optimize system performance.

### NOMENCLATURE

$\dot{m}_{inj}$	Mass flow rate in the injection line at pressure $p_{inj}[i]$	(kg/s)
$\dot{m}_{comp}$	Mass flow rate exiting the compressor at pressure $p_{inj}[i]$	(kg/s)
$\dot{m}_{cond}$	Mass flow rate through the condenser	(kg/s)
$p_{inj}[i]$	Pressure in the injection line	(kPa)

#### Greek Symbols

$\eta_s$	Isentropic efficiency of the compressor
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#### Subscripts

f	Saturated liquid	g	Saturated vapor
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