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*Board of Directors, International Institute of Ammonia Refrigeration*

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## Technical Paper #2

# Energy Consumption with CO<sub>2</sub>/Cascade Systems

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### **Abstract**

*CO<sub>2</sub> cascade systems have been used in refrigeration for many years and are growing in popularity for a number of reasons. Reduction in ammonia charge, reduced cost of refrigerant, reduction in compressor size, reduced risk of air ingress, and reduced energy consumption are all cited as possible advantages with CO<sub>2</sub> cascade. This study takes a detailed look at energy consumption using CO<sub>2</sub> cascade systems to determine whether they are in fact energy competitive with two stage ammonia systems or other possible systems over the range of temperatures normally encountered in food freezing and storage applications.*

*2009 IIAR Industrial Refrigeration Conference & Exhibition, Dallas, Texas*



## Introduction

The refrigeration and air-conditioning sectors are responsible for approximately 15% of all energy consumed worldwide, (IIR 2002). Improving the energy efficiency of refrigeration and A/C systems, would not only have a positive impact on the Earth's energy resources, but also would exert a positive effect on the indirect emission of CO<sub>2</sub>. The energy consumption of cooling systems is responsible for approximately 80% of the overall impact of the refrigeration sector on the greenhouse effect and global warming. The remaining 20% is from direct greenhouse gas emissions. Therefore, improving the energy efficiency in these systems is of significant importance to sustainability.

Much has been published in the last 10 years on the use of CO<sub>2</sub> in refrigeration systems. While offering the attraction of a natural working fluid with low toxicity, no flammability, and low direct Global Warming Potential, CO<sub>2</sub> still bears scrutiny for its cycle efficiency.

CO<sub>2</sub> has a relatively low critical temperature which means that it will not condense at temperatures above 87.8°F, (31°C). For most of the world, where condensing temperatures often exceed this level, the only option for using CO<sub>2</sub> in refrigeration is a cascade system; where the CO<sub>2</sub> is condensed in a heat exchanger against a different refrigerant with higher critical temperature, or a transcritical, (non-condensing) system. Though considered for vending machines, automotive AC and some small cooling applications, the cycle efficiency for transcritical systems is considerably worse than conventional systems and can hardly be seen as a “green” cooling option. Most applications considering transcritical CO<sub>2</sub> systems are niche markets where interest is based on something other than energy savings; (component size, high heat rejection temperature, etc.). That leaves cascade systems as the only realistic possibility for CO<sub>2</sub> in the vast majority of industrial systems where energy consumption is a significant portion of the total life cycle cost (Figure 1).

CO<sub>2</sub> cascade systems are certainly not new. Frick has records of many systems installed as early as the 1930s (Figure 2). Today, several hundred CO<sub>2</sub> cascade refrigeration systems are operating in industrial plants. With the recent renewed interest in CO<sub>2</sub> we have begun to see claims of 25–35% energy savings with CO<sub>2</sub> cascade systems when compared to direct two stage ammonia systems. Many of these claims are based on results from operating plants and the basis of the comparison is often not clear and difficult to determine. Comparative ratings developed from lab tests show far less optimistic efficiency for CO<sub>2</sub> cascade than is currently being claimed in our industry.

Our company offers a wide variety of refrigerant solutions and does not have a bias for or against CO<sub>2</sub> cascade systems. We do, however, believe that refrigerant choice should be based on the total assessment of the facts given a scientifically fact based treatment of alternatives. Numerous CO<sub>2</sub> cascade systems have been produced in the last 10 years using NH<sub>3</sub>, R-22, R-134a, and R-507 for the CO<sub>2</sub> condensing medium.

## **The Comparison**

The purpose of this paper is to show the results of a thermodynamic comparison of the cycle efficiency of a CO<sub>2</sub>/ammonia cascade system compared to a direct two stage ammonia system, then expand that analysis into real world efficiency comparisons. We believe this is a relevant comparison for most large food processing and cooling applications where two-stage ammonia is widely used today. Cycle efficiency calculations give a simple comparison of what is possible with any refrigerant between a given evaporator and condensing temperature. The comparison allows all assumptions to be known, and allows equal treatment of alternative systems. The resulting comparisons should be exactly the same regardless of who performs the calculations since the comparisons are only based on the known thermodynamic properties of the gases and are beyond question.

To understand the method, please review the calculations used for single stage compression shown on the Mollier diagram (Figure 3). For the simple cycle the refrigeration effect is only the enthalpy difference of the refrigerant entering and leaving the evaporator times the mass flow. For the initial cycle comparisons we assume 100% adiabatic efficiency of the compressor, so the power consumed is just the enthalpy difference of the refrigerant entering and leaving the compressor, (following a line of constant entropy), times mass flow. Since mass flow is the same through the evaporator and compressor we can ignore it, and the coefficient of performance, (COP), is just the evaporator enthalpy difference divided by the compressor enthalpy difference.

The same method can be used for two-stage compression (Figure 4). The capacity of the low stage is represented by the enthalpy difference of the refrigerant entering and leaving the low-stage evaporator, (650–100 Btu/lb = 550 Btu/lb). Low-stage power is enthalpy difference entering and leaving the low stage compressor, (725–650 Btu/lb). The capacity that the high-stage must handle is just the enthalpy difference of the refrigerant entering the low-stage evaporator and leaving the low stage compressor, (725–100 Btu/lb = 625 Btu/lb). This gives a multiplier for the capacity of the high stage compared to the low stage, which is used to increase the high-stage power by that multiple, (625/550 or 1.136 for this example). The two-stage COP is then (capacity of the low stage) / (low-stage plus high-stage power).

Performance of the cascade system follows the same logic with the only exception that the approach temperature in the cascade heat exchanger must be accounted for (Figure 5). We make the assumption that for any given CO<sub>2</sub> condensing temperature that the CO<sub>2</sub> side of the exchanger will be held to constant temperature. The high stage suction temperature will be lower than the CO<sub>2</sub> condensing by the amount of the heat exchanger approach temperature. For this comparison, (NH<sub>3</sub> two stage vs. CO<sub>2</sub> cascade), we assumed that the high stage of the cascade system is using ammonia as the refrigerant, so in effect the high side of both compared systems is

identical, except for the lower suction temperature required on the cascade system to overcome the approach temperature.

For the first comparison, compressor adiabatic efficiency was assumed to be 100% for all compressor stages. While not real world it shows that potential efficiency of the basic cycles. Figures 6 through 9 show the energy consumption of the CO<sub>2</sub> cascade system plotted against varying evaporator temperature as compared to a two stage ammonia system between the same temperatures. The plot on the left shows overall COP at varying evaporating temperatures and the curve on the right shows the percentage difference between the two curves on the left. The first four plots show the comparison as CO<sub>2</sub> cascade condensing temperature, and intermediate temperature on the two stage system is varied. This shows about a 2 to 6% energy penalty for the CO<sub>2</sub> cascade with the energy penalty growing with increasing intermediate temperature. These first four curves are calculated with a cascade approach temperature of zero; obviously impossible but providing a point of reference.

The next four curves, (Figures 10–13) show the impact on efficiency as the cascade approach temperature is increased from 1°F up to 9°F (5°C). Most of the systems being applied today have at least a 9°F approach temperature at full capacity because the cascade heat exchangers become significantly more expensive if designed for smaller approach temperature. Figure E4 demonstrates that at conditions of -40°F/ +20°F/95°F, CO<sub>2</sub> cascade has about 14.8% energy penalty over two-stage ammonia. Also, notice that the efficiency penalty with CO<sub>2</sub> cascade is almost the same from -60°F to -20°F taking 14–15% more energy than two-stage ammonia at all conditions.

To get back to real world systems, not perfect compressors, the adiabatic efficiency of real compressors operating at these conditions must be included to see how it affects the energy efficiency of the system. Using publicly available rating data from both reciprocating and screw compressors compatible with both ammonia and CO<sub>2</sub>, we



derived their adiabatic efficiency curves against discharge pressure and compression ratio. This has been incorporated into the same spreadsheet in order to see the effect on overall COP. Including real adiabatic efficiency increases the enthalpy difference of the gas entering and leaving the compressor at each operating condition. The consumed power increases for all compressor stages and results in more high stage load and power to account for the additional heat added to the refrigerant leaving the low stage compressor.

Figures 14 through 16 show this comparison using screw compressor efficiencies on all stages. The three curves show the efficiency results at three different heat exchanger approach temperatures. Looking at Figure 16, which is calculated with normal 9°F (5°C) cascade approach temperature, it shows that including real compressor efficiencies reduces the penalty with CO<sub>2</sub> cascade to break even at -60°F. At -40°F evaporating the penalty for using CO<sub>2</sub> cascade is still about 6% and increases to about 13% at -20°F (-28.9 C) evaporating. The reason for the reduction in the energy penalty for CO<sub>2</sub> cascade when real efficiencies are included is due to CO<sub>2</sub>'s higher density at lower suction temperature than ammonia. Frictional losses are a larger percentage of the total compressor power on the ammonia booster compressor operating on the much lighter ammonia gas. This gives support to the claim that the low stage efficiency is better on the CO<sub>2</sub> compressors at low evaporating temperature. However, there is no evaporating temperature above -60°F (-51°C) in this comparison where CO<sub>2</sub> cascade ever has an energy advantage over two stage ammonia when the normal 9°F (5°C) approach temperature of the cascade cooler and the required high stage power is included in the comparison.

## **Conclusion**

Cascade CO<sub>2</sub> averages 6% energy penalty at -40°F evaporating temperature over two-stage ammonia, with screw compressor efficiencies included in all stages.

Some claim that reciprocating compressors have much higher efficiency on CO<sub>2</sub> than screw compressors and this is responsible for the claimed energy advantage with CO<sub>2</sub> cascade. The next comparison takes the industry leading reciprocating compressor published efficiency for the CO<sub>2</sub> low stage and uses screw compressor efficiencies on the ammonia low and high stage to see if this is true (Figure 17). The CO<sub>2</sub> system still shows an energy penalty of about 1% at -60°F (-51°C) evaporating temperature, 8% at -40°F (-40°C) evaporating temperature, and climbs to about 12% penalty at -20°F (-28.9°C) evaporating temperature. Surprised at how closely this compared to the screw compressor efficiency data, the adiabatic efficiency of a common screw compressor against two reciprocating compressors designed for CO<sub>2</sub> duty were plotted in Figure 18. The efficiency curves are almost identical. While we have heard heady claims for improved efficiencies with reciprocating compressors on CO<sub>2</sub>, these claims are not supported by published ratings. Conclusion: Cascade CO<sub>2</sub> with reciprocating efficiency averages an 8% energy penalty.

Do CO<sub>2</sub> cascade systems demonstrate energy savings under any scenario? To answer this question a comparison was made using perfect compressors of 100% adiabatic efficiency on the CO<sub>2</sub> low stage and comparing to real screw compressor ratings available commercially today on the ammonia low stage and high stage (Figure 19). Even with perfect, totally loss-less CO<sub>2</sub> compressors against real ammonia screws on the low stage of the two stage ammonia system, the break-even temperature where CO<sub>2</sub> cascade starts to use less energy is still below -27°F (-33°C) evaporating temperature. With this comparison designed to show the maximum possible performance for CO<sub>2</sub>, the CO<sub>2</sub> cascade shows a 5% energy advantage at -40°F (-40°C). It is not possible to make CO<sub>2</sub> look any better than this. In the real world, no compressor will ever be 100% efficient. The best compressors made, whether reciprocating, screw or centrifugal do not exceed about 83% adiabatic efficiency when new at their optimum size, and speed. In fact, the best that could realistically be hoped for with any CO<sub>2</sub> cascade system, even if the compressors were operating at the optimum conditions to achieve 83% adiabatic efficiency, and new, would be 17% worse on the CO<sub>2</sub> compressors, and the system break-even point where CO<sub>2</sub> efficiency

would equal two stage ammonia would be below  $-50^{\circ}\text{F}$  ( $-46^{\circ}\text{C}$ ). One-hundred percent efficient compressors cannot overcome the efficiency disadvantage of CO<sub>2</sub> cascade (Figure 20).

At  $-20^{\circ}\text{F}/+20^{\circ}\text{F}/95^{\circ}\text{F}$ , ( $-29^{\circ}\text{C}/-6.7^{\circ}\text{C}/35^{\circ}\text{C}$ ), a condition where several recent CO<sub>2</sub> systems have been installed and are operating, only 32% of the total compressor power is consumed in the low stage CO<sub>2</sub> compressors, the remaining 68% is consumed in the high stage ammonia compressors. In order to save 32% power in these systems, (claims that have been made), the CO<sub>2</sub> compressors would have to consume ZERO POWER. In order to save 20% in total power, the CO<sub>2</sub> compressors would have to consume one third of their rated power. Again, this is clearly impossible.

The conclusion from the study of cycle efficiency of CO<sub>2</sub> cascade vs. two stage ammonia is that energy saving with CO<sub>2</sub> cascade is not possible given the fundamental thermodynamic disadvantage of the CO<sub>2</sub> cycle. It is possible to add complexity to both of the cycles that would improve overall efficiency, for example, both low and high stages could add economizer cycles and gain some efficiency improvement from liquid sub-cooling; however, to keep the comparison valid, these added cost options could be added to both systems and improve the COP of both cycles. In a valid comparison, adding complexity does not give an added advantage to the CO<sub>2</sub> systems.

Some have argued that CO<sub>2</sub> is less sensitive to pressure drop, so allowing the same pressure drop in the ammonia low stage as the CO<sub>2</sub> low stage gives CO<sub>2</sub> a large advantage. This argument is true, but a responsible system designer would design an ammonia low stage with such excessive pressure drop thus giving away 20 or 30% energy. While there are probably many ammonia systems in operation that are far from optimum, and could benefit from attention to details in the system design, and actual operation, the same could be said for CO<sub>2</sub> systems if they were used in the same numbers as two stage ammonia systems.

Arguments have been presented for cold storage facilities at  $-20^{\circ}\text{F}$  ( $-28.9^{\circ}\text{C}$ ) that show the advantage of  $\text{CO}_2$  reciprocating compressors against ammonia two stage screw compressors is because the reciprocating compressors have better part load performance, and because the systems operate at part load most of the time. First, booster compressors operate at low compression ratios, where conventional slide valve unloading of screw compressors is very efficient. Efficient plants use sequencing on all their compressors and turn off machines not needed, while trimming capacity with one compressor. Most modern ammonia plants use different size compressors (Figure 21), so it is never necessary to run a compressor far unloaded for an extended period of time. With the  $\text{CO}_2$  cascade system starting from a 12% energy penalty at a  $-20^{\circ}\text{F}$  ( $-28.9^{\circ}\text{C}$ ) evaporating temperature it would be necessary to improve the part load efficiency of the  $\text{CO}_2$  boosters by about 36% in order to offset the 12% penalty of the cycle, just to get back to breakeven. This level of improvement is certainly not supported by published part load data on reciprocating compressors.

Some plants concerned with optimizing part load power on screws that need to run unloaded for long time periods choose variable speed drives. The VSD gives a significant boost in the compressor part load performance and can easily improve upon reciprocating COP at part load.

On the plus side,  $\text{CO}_2$  cascade systems have many advantages in industrial refrigeration. Reduction in size of the low stage compressors and reduction of the pipe sizes to the low side are beyond dispute. Using  $\text{CO}_2$  as a direct refrigerant or as a heat transfer fluid into food storage areas certainly allows the possibility to significantly reduce the ammonia charge. Whether a  $\text{CO}_2$  leak into a storage area actually improves safety over an ammonia leak into a storage area is still a subject for separate debate.

$\text{CO}_2$  systems always operate in positive pressure, reducing the risk of air and moisture ingress. However, due to the much greater problems associated with water

contamination in CO<sub>2</sub> systems, servicing techniques are much more important and quite different than current ammonia practice. Also, of concern is that typical CO<sub>2</sub> oils in use are hygroscopic and absorb water if left open at atmospheric pressure. Again, this just means that training for operation and servicing of CO<sub>2</sub> systems is critical to success.

A comparative analysis of a 150,000 sq. ft. (14,000m<sup>2</sup>) refrigerated distribution facility estimated total energy usage for refrigeration of 5,021,000 kWh per year. (Gooseff and Horton, 2008). The portion of the energy usage for the -10°F (12°C) freezer was 2,563,000 kWh per year. If this facility chose CO<sub>2</sub> Cascade instead of two stage ammonia for the freezer portion of the load at -20°F (7°C) saturated suction, the energy penalty for CO<sub>2</sub> cascade would be \$32,000 per year or 228 metric tons of equivalent CO<sub>2</sub> emission. Over an estimated 20-year life of the plant, the additional energy cost would be \$640,000, and the CO<sub>2</sub> emission penalty would be over 4500 metric tons (Figure 22).

There is sufficient merit for CO<sub>2</sub> cascade that it can be the best choice in some low temperature applications where the benefits outweigh the disadvantages, however, energy savings is not the reason to choose CO<sub>2</sub> cascade over direct two-stage ammonia as it will almost always require additional power consumption.

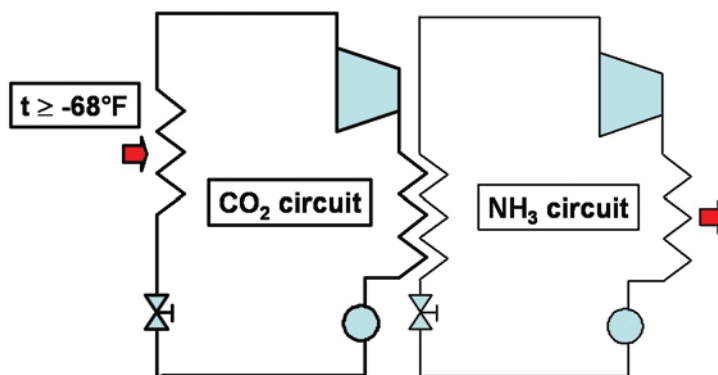
In summary, CO<sub>2</sub> Cascade's low cycle efficiency, and the need for the cascade condenser with its unavoidable approach temperature, cause increased power consumption and a reduction in overall COP that is only partially offset by the improved compressor efficiency on the CO<sub>2</sub> low-stage compressors. Some applications may warrant paying the higher energy costs.

## References

IIR, 2002, *Report on refrigeration sector achievements and challenges, The World Summit on Sustainable Development*, Johannesburg, South Africa.

*Ammonia as the Sustainable Refrigerant: An ammonia-Halocarbon Comparison*, Gooseff and Horton, Tech Paper 3, IAR Conference March 2008.

Figure 1. Carbon Dioxide/Ammonia Cascade Systems

Figure 2. Example of CO<sub>2</sub>/Cascade Article from 1932 *Power Magazine*

## Advantages of CO<sub>2</sub>-Ammonia System For Low-Temperature Refrigeration

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ANYONE familiar with carbon dioxide knows that it evaporates at extremely low temperatures at suction pressures much above atmospheric. This characteristic led to the application of CO<sub>2</sub> to refrigerating purposes at low temperatures, but the extremely high condensing pressures usually encountered discouraged the adoption of either the straight or the compound CO<sub>2</sub> compression systems for low-temperature quick freezing plants. It was then discovered that evaporating ammonia to condense the CO<sub>2</sub> gas made the operation more feasible. Naturally the low head pressures under which the CO<sub>2</sub> operates in this type of system has encouraged still lower suction pressures, with lower compression ratios than are found in compound-compression plants employing this gas.

The production of low temperatures, economically, has become a topic of intense interest to the refrigeration industry, especially since the advent of quick-frozen foods. Of the various methods tried, the split-stage system has attracted most attention. This system has played an important rôle in the freezing and preservation of fruits, meats, ice creams, fish and other foods, and is being extended into other fields, such as process work in refineries, research work in laboratories and experimental work in industrial plants

The split-stage system outlined in Fig. 2 operates as two distinct refrigeration systems. The low-pressure CO<sub>2</sub> gas is drawn into the CO<sub>2</sub> compressor and is discharged into the condensers, which are cooled by direct expansion of ammonia. The suction of the ammonia compressor removes the ammonia gas evaporated in cooling the CO<sub>2</sub> and discharges it into the usual ammonia high side. The ammonia liquid level in the CO<sub>2</sub> condensers is maintained by float control, thus insuring flooded conditions and a relatively low temperature difference between the CO<sub>2</sub> and the ammonia. This difference depends on the amount of surface, but can be maintained between 10 and 15 deg. F. with 8½ sq.ft. of surface per ton of refrigeration when the equipment is properly designed.

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POWER—January 19, 1932

**Figure 3. Simple Refrigeration Cycle Showing COP Calculation**

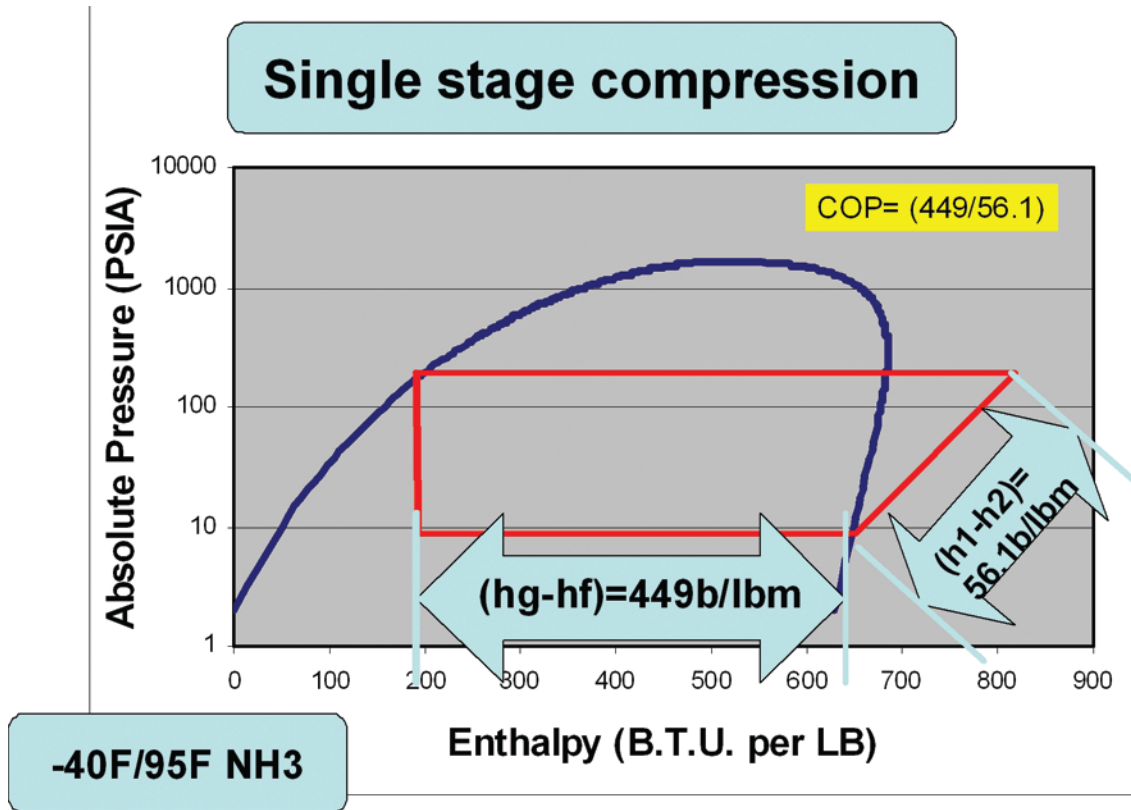




Figure 4. Two Stage Simple Cycle

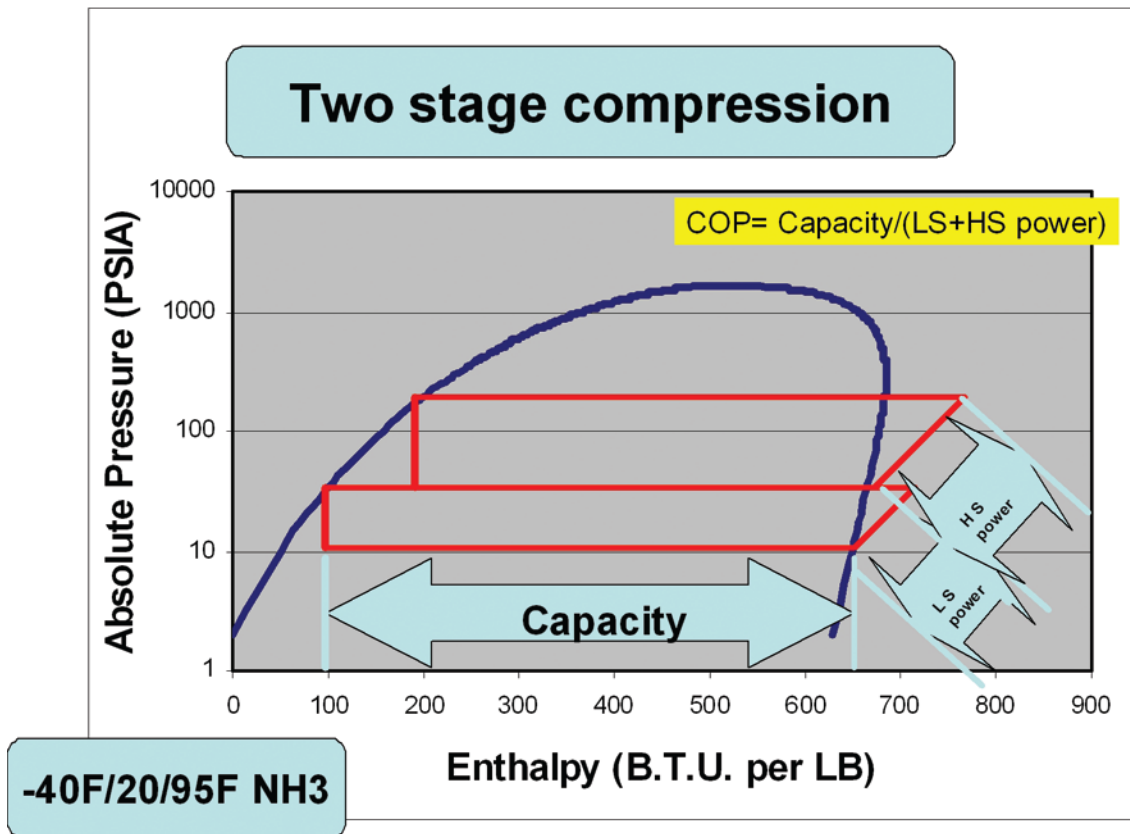
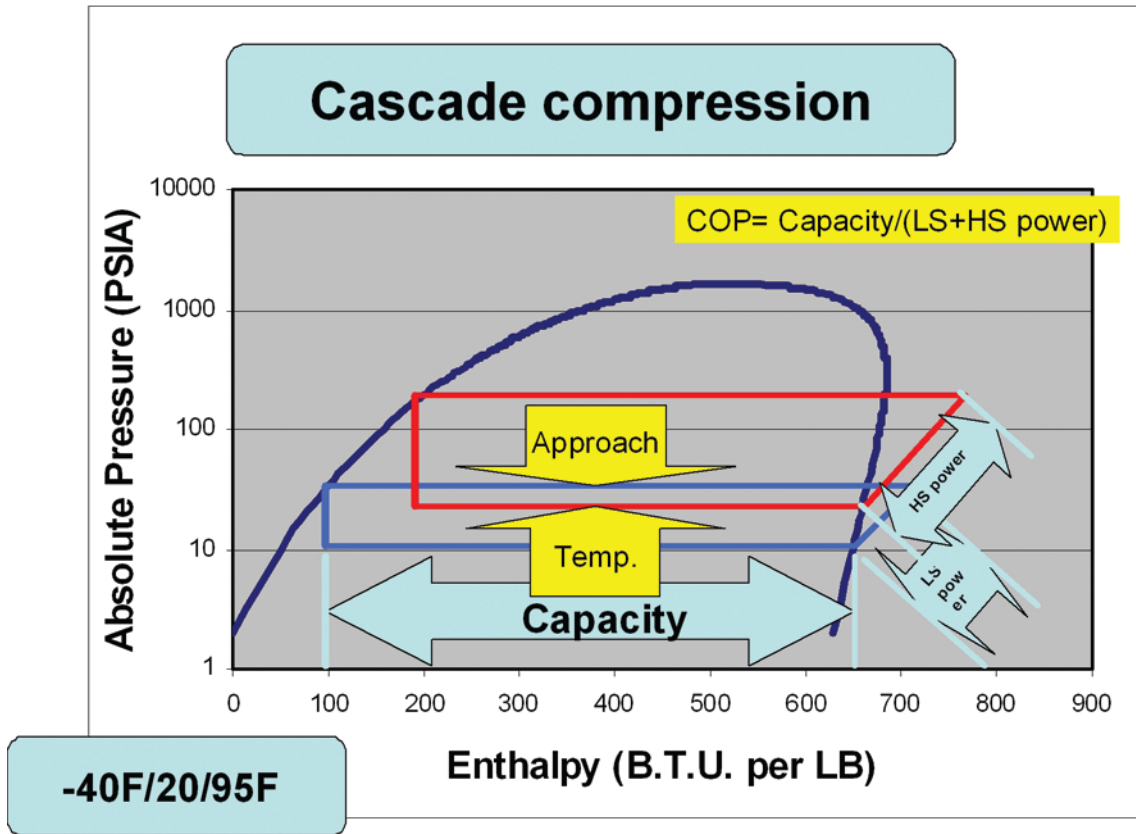
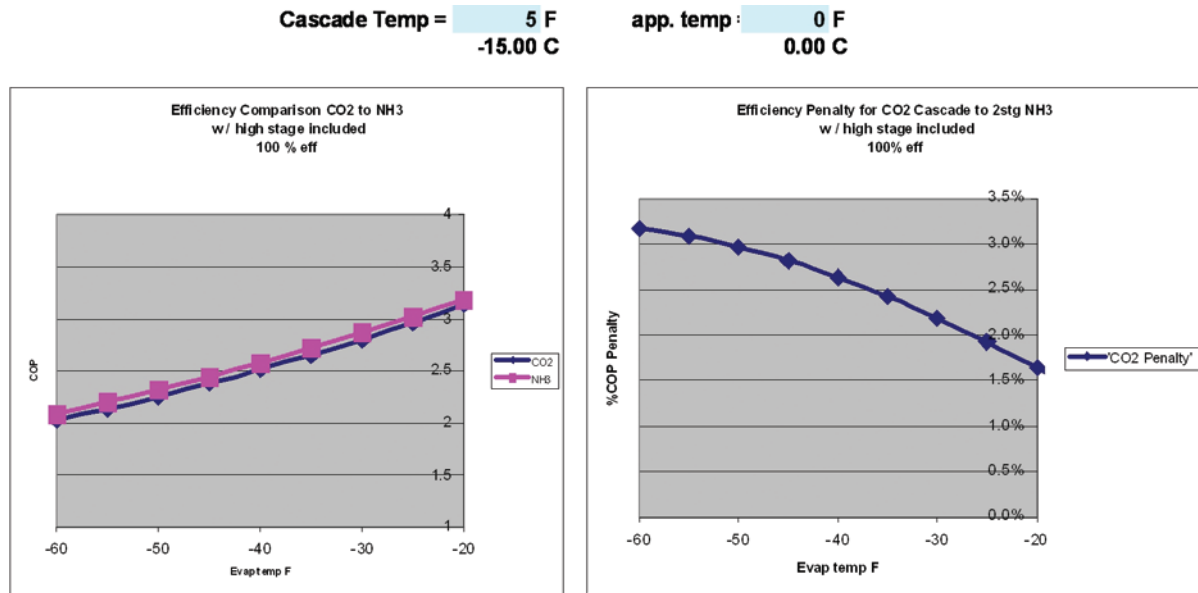


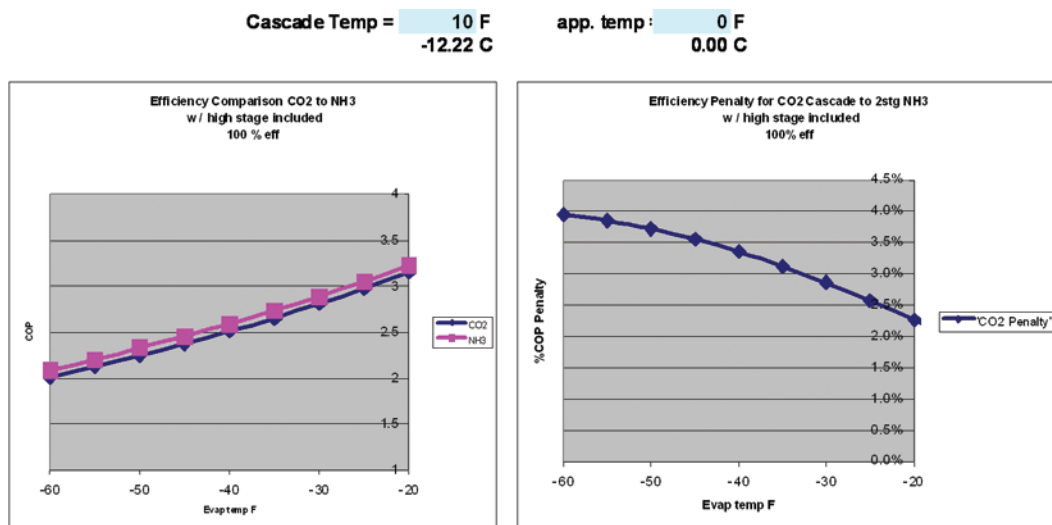
Figure 5. Cascade Simple Cycle



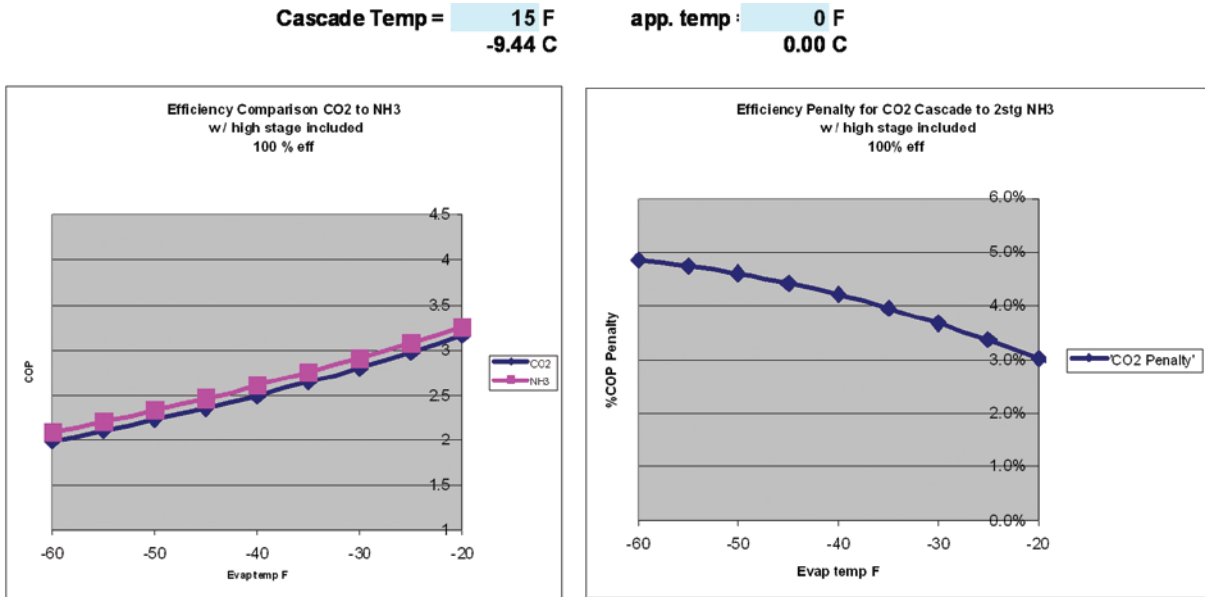
**Figure 6. Results of Cycle Calculations Showing CO<sub>2</sub> Cascade Penalty at 5°F Cascade Temperature and 0° Approach Temperature**



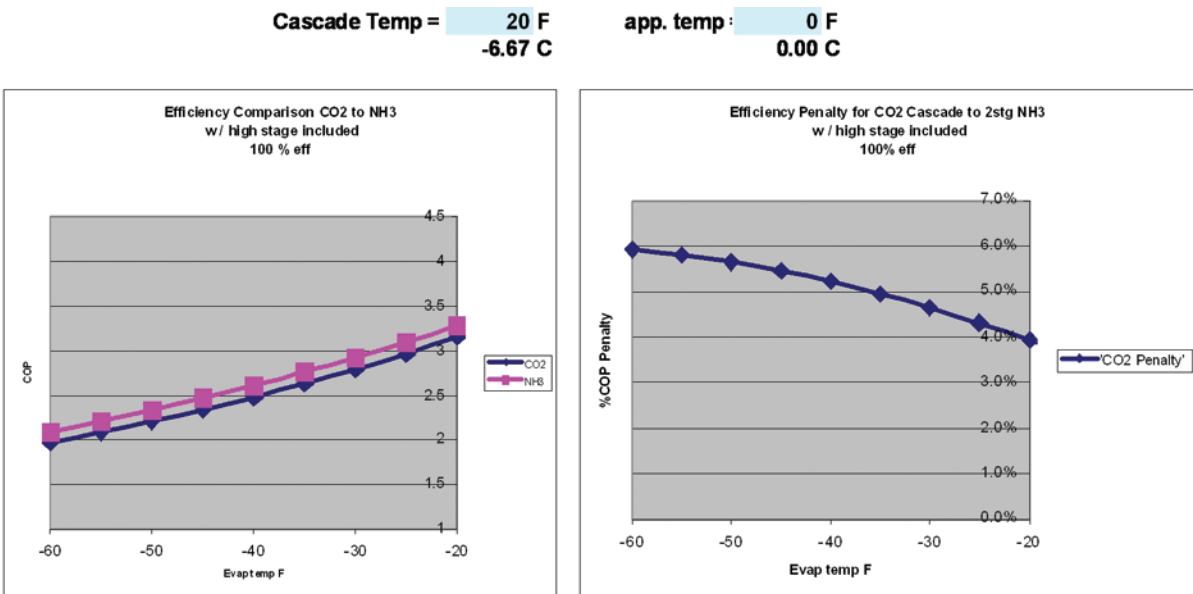
**Figure 7. Results of Cycle Calculations Showing CO<sub>2</sub> Cascade Penalty at 10°F Cascade Temperature and 0° Approach Temperature**



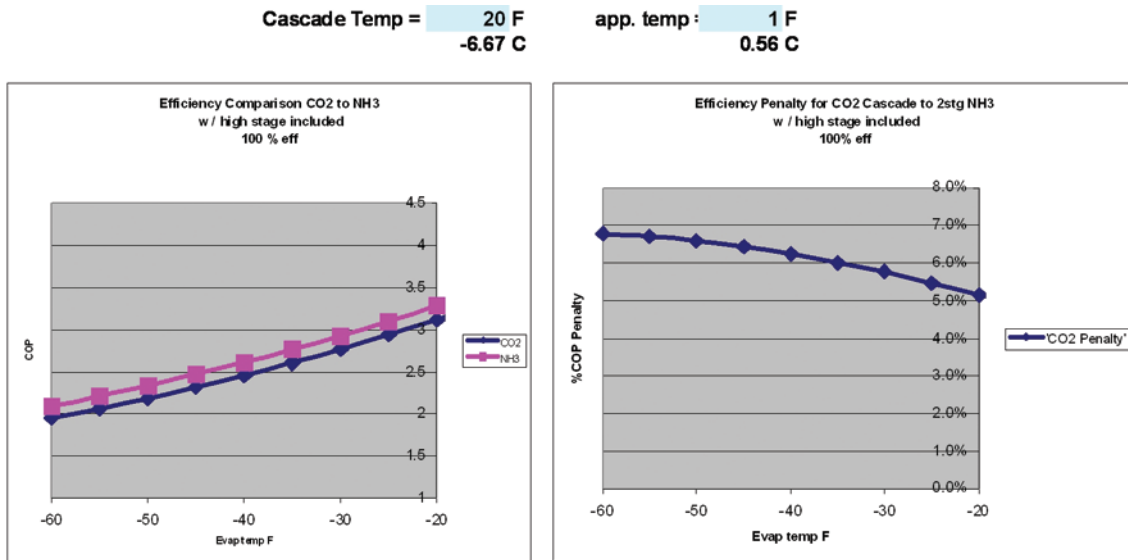
**Figure 8. Results of Cycle Calculations Showing CO<sub>2</sub> Cascade Penalty at 15°F Cascade Temperature and 0° Approach Temperature**



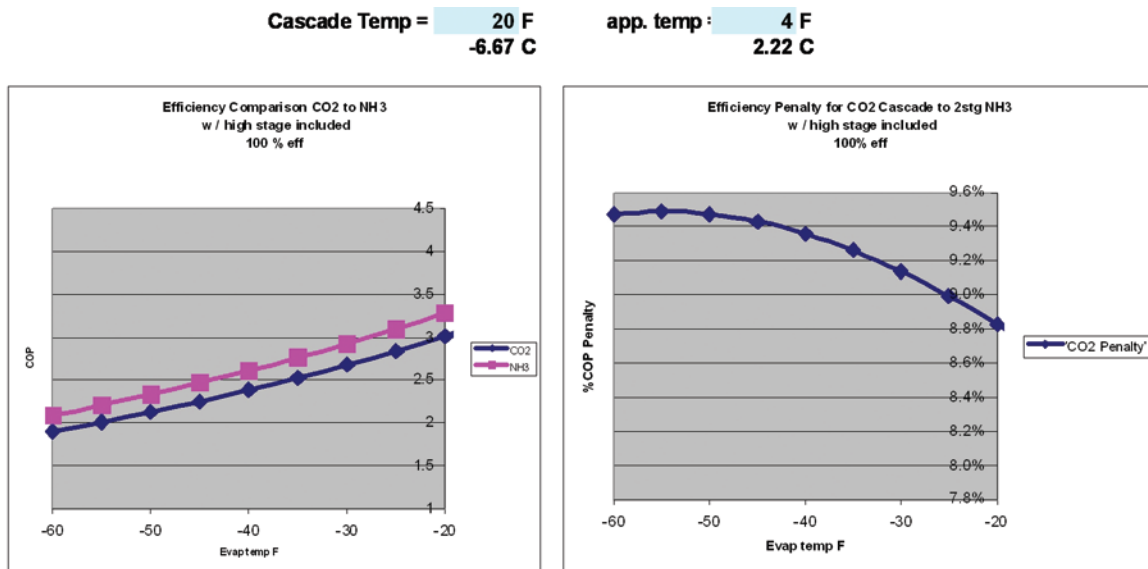
**Figure 9. Results of Cycle Calculations Showing CO<sub>2</sub> Cascade Penalty at 20°F Cascade Temperature and 0° Approach Temperature**



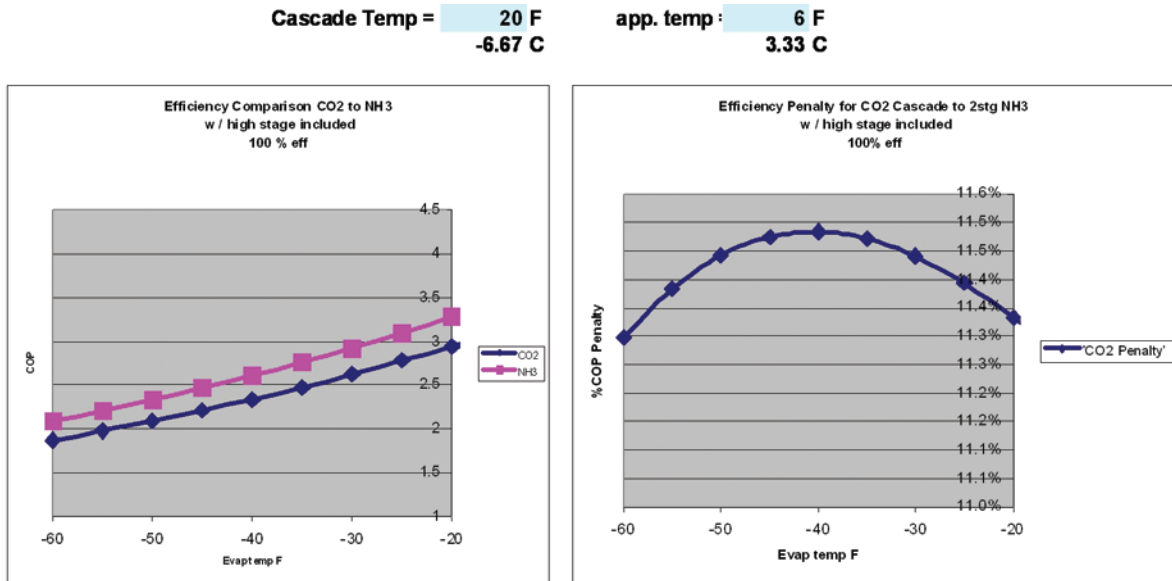
**Figure 10. Result of Cycle Calculations Showing Impact of Increasing Approach Temperature (1°F)**



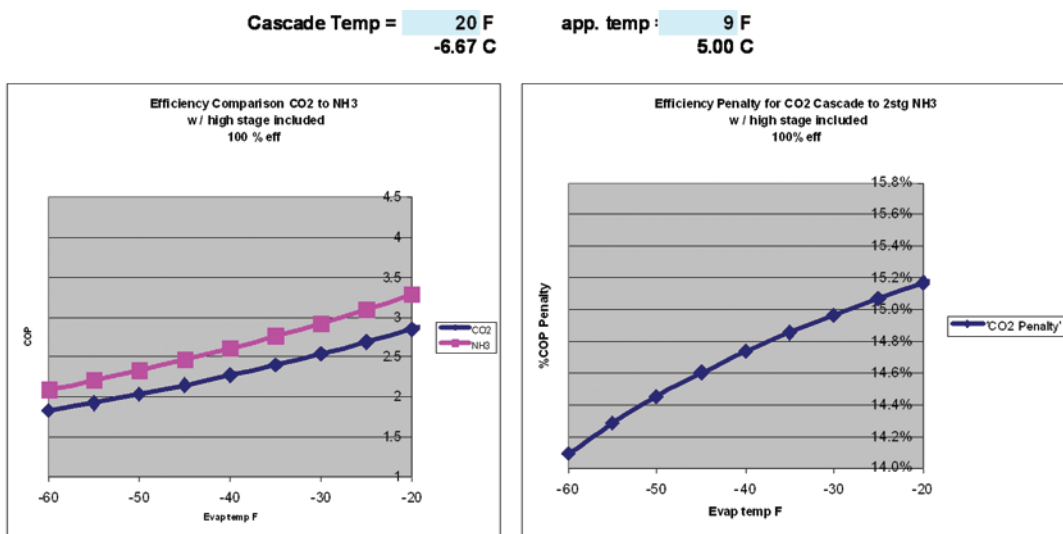
**Figure 11. Result of Cycle Calculations Showing Impact of Increasing Approach Temperature (4°F)**



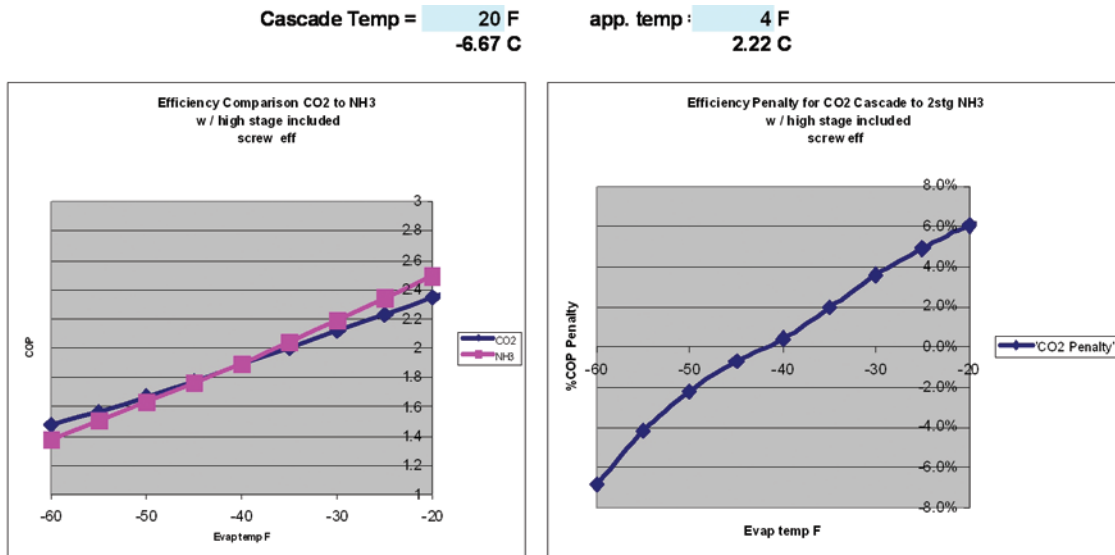
**Figure 12. Results of Cycle Calculations Showing Impact of Increasing Approach Temperature (6°F)**



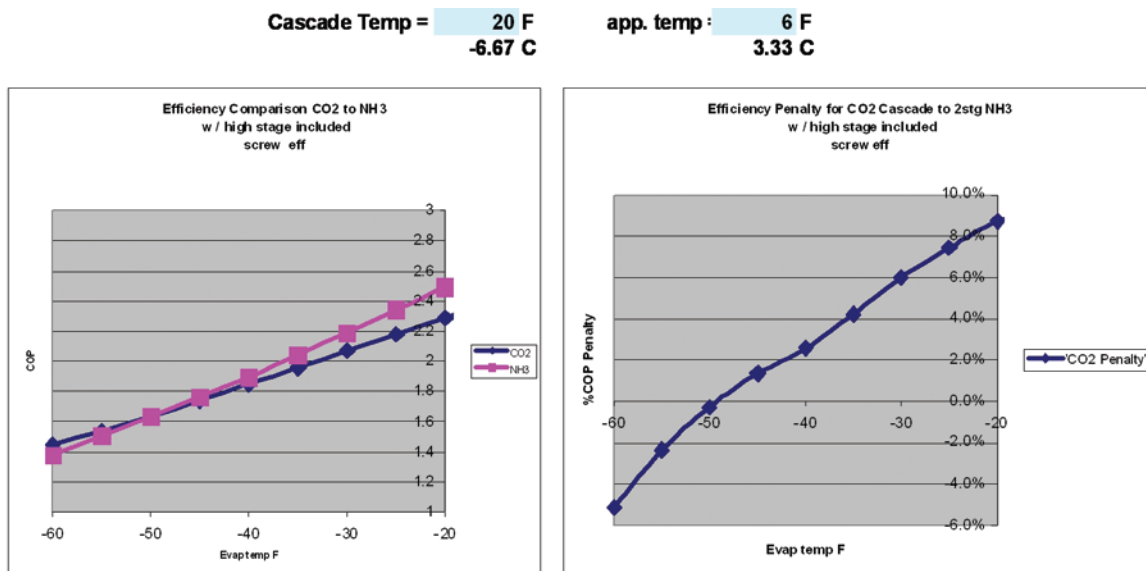
**Figure 13. Results of Cycle Calculations Showing Impact of Increasing Approach Temperature (9°F)**



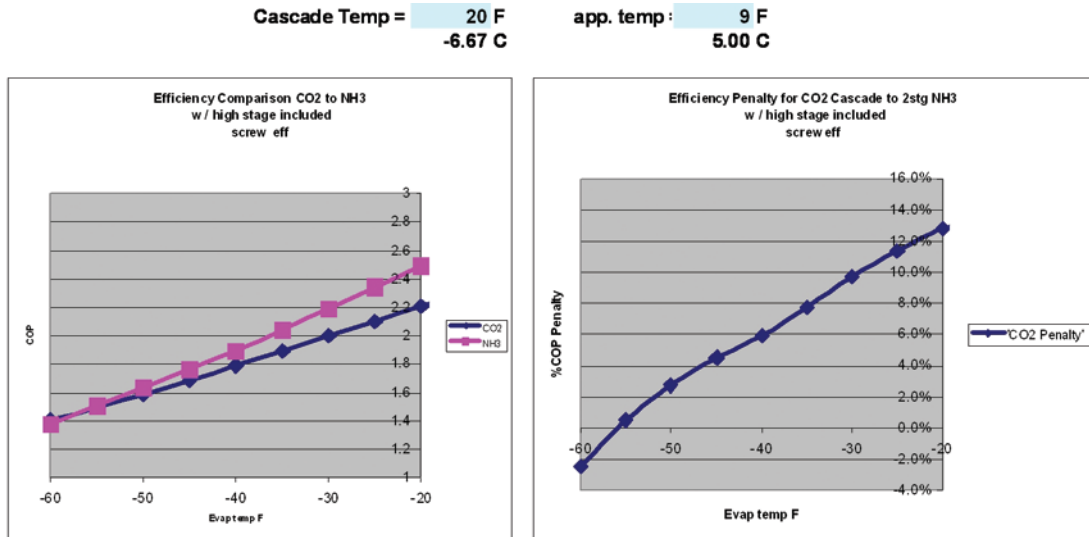
**Figure 14. Results of Cycle Calculations with Screw Compressor Efficiencies on All Stages, 4°F Approach Temperature**



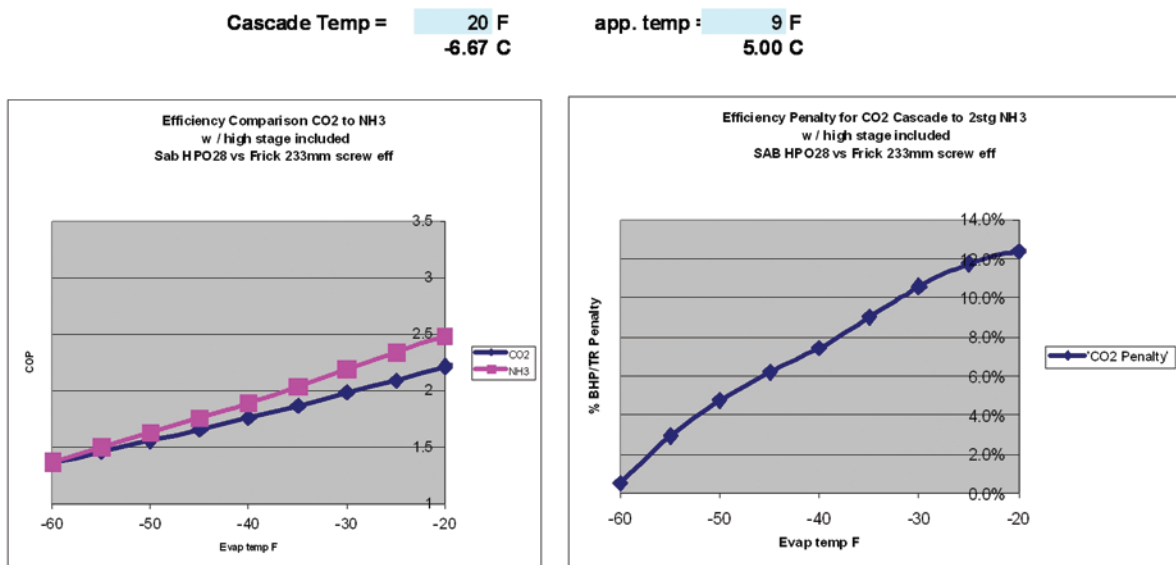
**Figure 15. Results of Cycle Calculations with Screw Compressor Efficiencies on All Stages, 6°F Approach Temperature**



**Figure 16. Results of Cycle Calculations with Screw Compressor Efficiencies on All Stages, 9°F Approach Temperature**

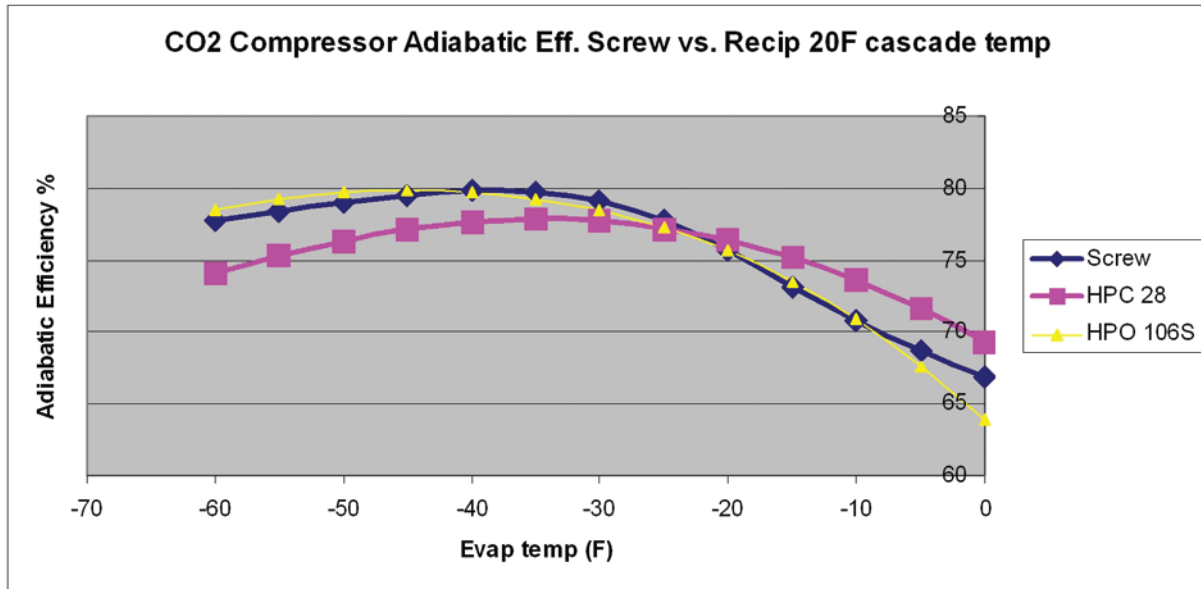


**Figure 17. Reciprocating Efficiency on CO<sub>2</sub> Compressors vs. Screw Efficiency on NH<sub>3</sub> Two Stage**

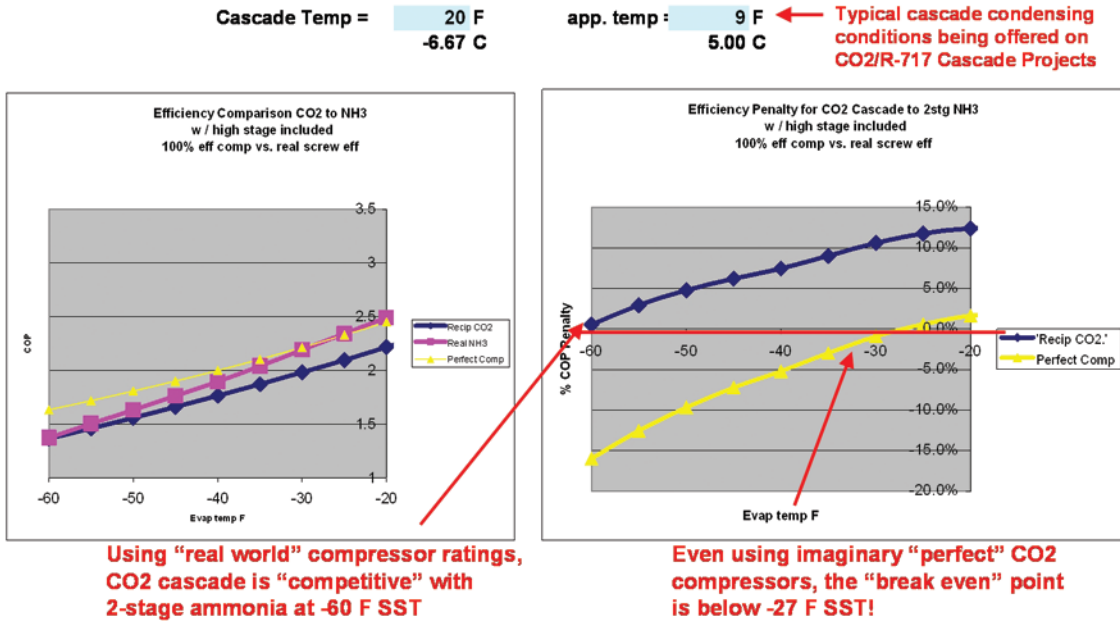




**Figure 18. Comparison of Adiabatic Efficiency of Two CO<sub>2</sub> Reciprocating Compressors and One Screw Compressor**

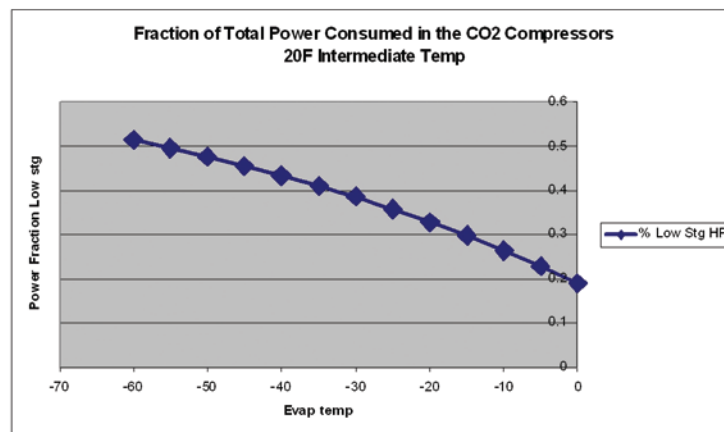


**Figure 19. If CO<sub>2</sub> compressors were 100% efficient compared to real ammonia two stage ratings, CO<sub>2</sub> cascade would offer energy savings only below -27°F (-33°C) evaporating temperature**



**Figure 20. Fraction of Total Power Consumed in the CO<sub>2</sub> Compressors**

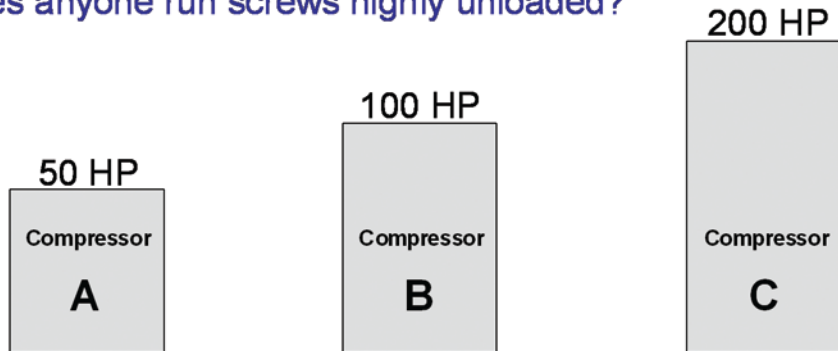
At -20/+20/95F, CO<sub>2</sub> Cascade, 32% of the total compression power is consumed in the CO<sub>2</sub> compressors



Conclusion : In order to save 32% in energy, the CO<sub>2</sub> compressors would have to consume ZERO power..! In order to save 20% energy the CO<sub>2</sub> compressors have to consume one third of rated power. NOT POSSIBLE

**Figure 21. Multiple Compressor Strategies Avoid Part Load Penalties**

Multiple compressor strategies avoid part load penalties –  
Does anyone run screws highly unloaded?



- |                 |            |           |
|-----------------|------------|-----------|
| 1. 0 - 50 HP    | Compressor | A         |
| 2. 50 - 100 HP  | Compressor | B         |
| 3. 100 - 150 HP | Compressor | A & B     |
| 4. 150 - 200 HP | Compressor | C         |
| 5. 200 - 250 HP | Compressor | A & C     |
| 6. 250 - 300 HP | Compressor | B & C     |
| 7. 300 - 350 HP | Compressor | A & B & C |

**Figure 22. Calculation of CO<sub>2</sub> Energy Penalty and CO<sub>2</sub> Equivalent Emission for -20°F Freezer**

CO <sub>2</sub> Cascade Energy Penalty @ -20F (-28C) Sat Suct		
<b>Per year</b>	2,563,209	kWH Total Freezer energy consumption per year
Penalty for CO <sub>2</sub> cascade vs. 2 stg NH <sub>3</sub>	12.50%	percentage energy penalty
Excess energy consumption per year	320,401	kWH Freezer energy penalty per year
Excess energy cost @ \$0.10/kWH	\$32,040	
Metric tons of Carbon Dioxide equivalent	228	metric Tons CO <sub>2</sub> emission per year
<b>Per 20 year life</b>		
Excess energy consumption per year	6,408,023	kWH
Excess energy cost @ \$0.10/kWH	\$640,802	US Dollars
Metric tons of Carbon Dioxide equivalent	4,567	metric Tons CO <sub>2</sub> emission
per 20 year life due to energy penalty		per 20 year life due to energy penalty

