Energy Efficient Design, Retrofit and Control of Evaporative Condensers in Ammonia Refrigeration Systems

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ABSTRACT

Ammonia refrigeration systems typically offer many energy efficiency opportunities because of their large power-draw, long runtimes and dynamic operation. Ammonia refrigeration system energy use is highly dependent upon condensing head pressure, which in turn is a function of evaporative condenser capacity and control. This paper investigates the relationship between system energy use, condenser capacity and condenser control. It begins by developing a methodology to determine condenser performance and then to simulate the annual energy use of the compressors and condenser fans. The simulation model, which uses both engineering fundamentals and empirical data, accurately captures synergistic effects between the compressors, the condenser and ambient wet-bulb temperature. Energy savings are then calculated for three cases: installing VFDs on condenser fans, using wet-bulb approach control strategy and increasing condenser performance. To illustrate the effect of climate, these simulations are performed for two different ASHRAE climate zones, Miami, FL and Minneapolis, MN, which are hot and cold climates respectively. The results show that improving the performance of an underperforming condenser is the most cost effective energy conservation measure irrespective of location. However energy savings from installing VFDs on condenser fans and utilizing wet-bulb approach strategy depends on the ambient weather conditions. Next, the internal rate of return is calculated for installing additional condenser capacity beyond standard practices in new construction applications for the same two ASHRAE climate zones. The results indicate that the internal rate of return exceeds 20% for installing twice the baseline condenser capacity. In summary, the paper presents an integrated approach for design, retrofit and control of evaporative condensers in ammonia refrigeration systems. Energy savings derived through the use of this approach can significantly improve energy efficiency of ammonia refrigeration systems.

Introduction

About 7.5% of the total manufacturing energy consumption is used by the food processing industry and about 21% of this energy is electricity (EIA 2006). In many of these facilities, the ammonia refrigeration system is the largest energy consumer. Refrigeration and process cooling account for 27% of the electricity use in the food processing sector (EIA 2006). Refrigeration system energy use is highly dependent upon condensing pressure, which in turn is a function of condenser capacity and control. Thus, improving condenser capacity and control can lead to significant energy savings.

The paper begins by determining actual condenser performance using data from a refrigeration control system. A simulation model, which uses the condenser performance, is then developed to calculate annual energy use of the compressors and condenser fan under study. The simulation model is then used to calculate energy savings for three energy conservation measures

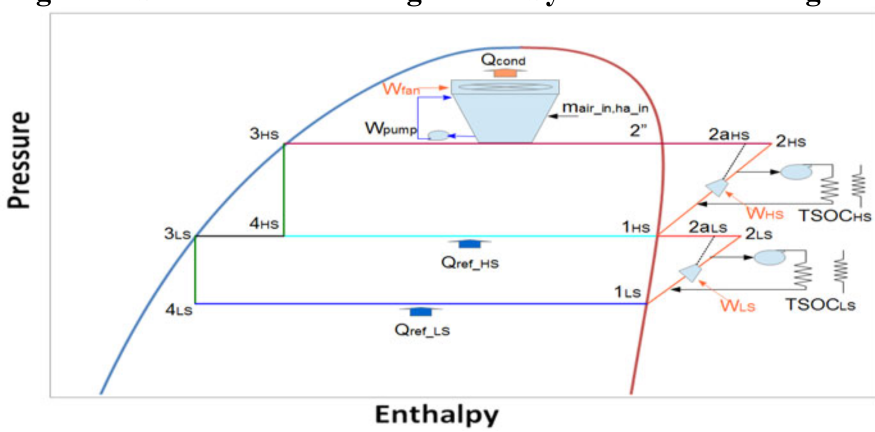
(ECMs): installing a VFD on the condenser fan, using wet-bulb approach strategy and increasing condenser performance. To illustrate the effect of climate, these simulations are performed for Miami, FL and Minneapolis, MN, which are hot and cold climates respectively. Finally the paper determines the internal rate of return for installing additional capacity beyond standard practices in new construction applications.

System Description

The system analyzed is a two-stage ammonia refrigeration system with two low-stage compressors and two high-stage compressors. All the compressors are screw type with slide-valve control and thermo-syphon oil cooling. One evaporative condenser with a constant speed fan rejects heat from the system. For the remainder of this paper, the term system will refer to the condenser fan and compressors. Energy use from the condenser pump is small and is not evaluated in this paper.

Key system parameters including motor current, ammonia pressures and temperatures were obtained from the refrigeration control system. Property data for ammonia was calculated using the Reference Fluid Thermodynamic and Transport Properties Database (NIST, 2010) also known as REFPROP. Figure shows a schematic of the refrigeration system understudy.

Figure 1. Schematic for Refrigeration System on a P-H diagram



Calculating Heat Rejection to Condenser

Condensing pressure is a key variable that determines system energy use. To accurately calculate condensing pressure, the condenser performance must be determined. The first step in determining condenser performance is to calculate the total heat rejected from the compressors to the condenser. An energy balance on the system shows that the total heat rejected by the condenser is the refrigeration (Qref) provided by both the low and high-stage compressors plus the heat of compression or shaft power (Ws) of both the low and high-stage compressors.

$$Q_{Cond.actual} = \sum Qref_{LS} + \sum Qref_{HS} + \sum Ws_{LS} + \sum Ws_{HS} \quad (1)$$

All the heat reject rejected from the low-temperature compressors minus the thermo-syphon oil cooling ($TSOC_{,LS}$) of the low-stage compressors is transferred to the high-stage system. Thus, the refrigeration provided ($TR_{provided,HS}$) by the high-temperature compressors is:

$$\sum TR_{provided,HS} = \sum Qref_{LS} + \sum Qref_{HS} + \sum W_{LS} - \sum TSOC_{,LS} \quad (2)$$

The condensers must reject the heat from the refrigeration and shaft power of the high-stage compressors, plus the TSOC from the low-stage compressors. Thus the heat rejection to the condenser consists of three principle components:

$$Q_{Cond.actual} = \sum TR_{provided,HS} + \sum Ws_{HS} + \sum TSOC_{,LS} \quad (3)$$

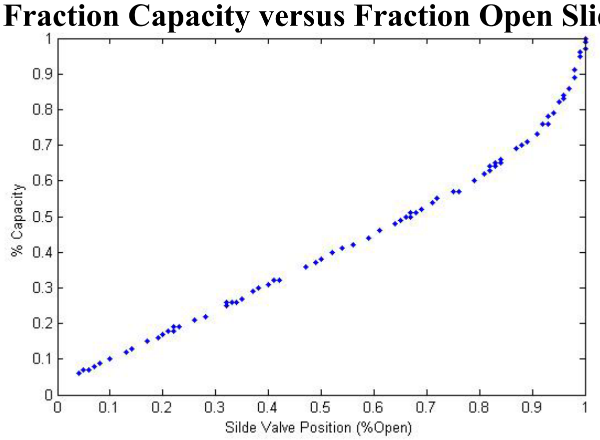
Refrigeration Provided by the High Stage Compressors ($TR_{provided,HS}$)

Manufacturers' provide the rated refrigeration capacity or tons rated (TR) of a compressor for a range of suction and condensing temperatures. This data can be fitted to a second order polynomial equation with interaction terms (Manske 2000) to determine the rated capacity at a given suction and condensing temperature as:

$$TR = C_0 + C_1 \cdot T_{cond} + C_2 \cdot T_{suc} + C_{11} \cdot T_{cond}^2 + C_{22} \cdot T_{suc}^2 + C_{12} \cdot T_{cond} \cdot T_{suc} \quad (4)$$

The percent refrigeration capacity of a compressor is a function of the slide valve percent open position, but they are not identical. In this case, percent refrigeration capacity as a function of the slide valve position is shown in Figure 2. The data were obtained from the refrigeration control system. The curve is similar to the one reported by Stoecker (1998).

Figure 2. Fraction Capacity versus Fraction Open Slide valve position



The suction and condensing temperatures are determined using the respective pressures and ammonia property data. The actual refrigeration capacity of the compressor can be calculated using Equation 4 and percent refrigeration capacity from the control system, as:

$$Q_{ref} = Qr_{rated} \cdot \%Capacity \quad (5)$$

Heat of Compression Produced by High Stage Compressors (Ws_{HS})

Motor current of each compressor was obtained from the control system data. To correlate motor current to shaft power (Ws), a relationship between motor current and input power must be developed. This relationship, f(A), can be developed from point measurements of motor current and input power throughout the operating range of the compressor. Through the use of both the name-plate efficiency of the compressor ($\dot{\eta}_m$) and f(A), shaft power or equivalently the heat of compression of each compressor can be calculated as:

$$Ws_{HS} = f(A) * \dot{\eta}_m \tag{6}$$

Thermosyphon oil cooling (TSOC). Consider the low-temperature cycle of the two-stage ammonia refrigeration system represented in Figure 1. At state 1_{LS}, ammonia enters the compressor as a saturated vapor and leaves the compressor as a super-heated vapor at state 2_{LS}. The path 1_{LS} - 2_{LS} represents the heat of compression. Ideally the heat rejected to the next stage would be the difference in enthalpy of points 2_{LS} and 3_{LS}. However, some heat is lost to the compressor oil and the actual temperature of the refrigerant at the exit of the compressor is at point $2a_{LS}$ rather than 2_{LS}. The temperature and pressure at point $2a_{LS}$ are known from the control system data and hence the enthalpy at point $2a_{LS}$ can be determined using property data for ammonia.

Heat of compression of the low-stage compressors, Ws_{LS}, can also be calculated using Equation 6. By applying an energy balance on the compressor discharge the heat transferred to the thermo-syphon oil cooling system can be calculated as:

$$TSOC_{,LS} = Ws_{LS} - m_{ref.LS} \cdot (h_{2a.LS} - h_{1.LS}) \tag{7}$$

The mass flow of refrigerant through the low-stage compressors can be calculated as:

$$m_{ref\text{-}LS} = Q_{ref.LS} / (h_{1.LS} - h_{4.LS}) \tag{8}$$

Determining Evaporative Condenser Performance

The effectiveness, eff, of an evaporative condenser can be modeled as (Manske, Reindl & Klein 2001):

$$\text{eff} = \frac{\text{Condenser Capacity}}{\text{Maximum Condenser Capacity}} = \frac{\dot{m}_{air} \cdot (h_{air,out} - h_{air,in})}{\dot{m}_{air} \cdot (h_{air,out}\|_{Trefrigerant,Sat} - h_{air,in})} \tag{9a,b}$$

Where $\dot{m}_{air}$ = mass flow rate of through evaporative condenser from manufacturers data

$h_{air,out}$ = enthalpy of air leaving evaporative condenser

$h_{air,in}$ = enthalpy of air entering evaporative condenser

$h_{air,out}\|_{Trefrigerant,Sat}$ = enthalpy of saturated air at condensing temperature

Typically, manufacturers' report volume flow rate of air, nominal capacity, and the heat rejection factor (HRF). The volume flow rate is used to calculate the mass flow rate using the density of air at standard conditions. The HRF, which is a function of both the outside air wet-bulb temperature (T_{wb}) and the saturated condensing temperature (T_{cond}), is used to determine the rated condenser capacity for a given T_{wb} and T_{cond} as (Manske, Reindl & Klein 2001):

$$\text{Rated Condenser Capacity} = \text{Nominal Capacity} / \text{HRF}\ (T_{wb}, T_{cond}) \qquad (10)$$

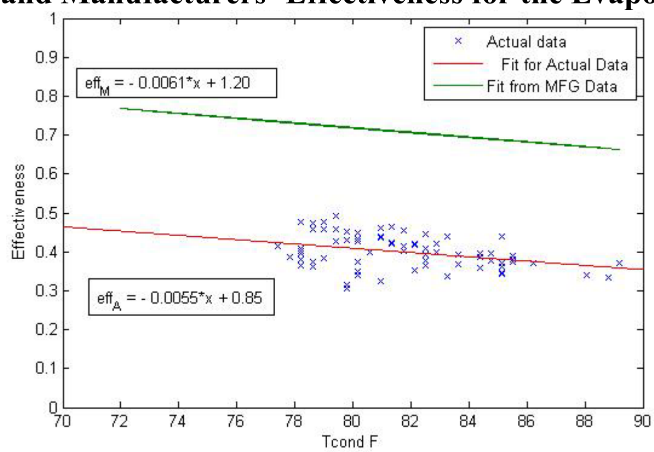
Equations 9b and 10 can be applied to manufacturers' specifications for an evaporative condenser to determine a relationship between T_{cond} and effectiveness for a given wet-bulb range. Effectiveness is found to be linearly related to T_{cond} as:

$$eff_M = e_0 - e_1 \cdot T_{cond} \qquad (11)$$

Since the actual capacity of the evaporative condenser during operation has been calculated, the actual effectiveness can be fit to a line in the form of Equation 11. Measured effectiveness versus T_{cond} data from the system under study are plotted when both the evaporative condenser fan and pumps are at full capacity in Figure 3. The manufacturer rated effectiveness from Equation 11 is also plotted on the same chart to compare the effectiveness of a new evaporative condenser to one which has been in service for a few years.

Figure 3 indicates that the evaporative condenser performance has degraded over time. The actual capacity is about 40% less than the manufacturer's rated capacity. This information can be used as a calibration parameter for simulation programs. For example, in Figure 3, the condenser capacity for a new condenser would be about 1.69 times the current actual capacity.

Figure 3. Actual and Manufacturers' Effectiveness for the Evaporative Condenser



Simulating Annual Energy Use

The annual energy use of a refrigeration system is the sum of compressor and condenser fan energy use. Condensing pressure is a key variable that must be calculated correctly to properly simulate compressor and condenser fan energy use. The following steps outline a methodology to calculate compressor power, condensing pressure and condenser fan power.

Calculating Compressor Input Power

Rated shaft power (bhp_{rated}) of a given compressor over a range of suction and condensing temperatures can be obtained from the manufacturer. This data can be fitted to a second-order polynomial equation with interaction terms to determine the rated full-load shaft power at given suction and condensing temperatures (Manske 2000), as:

$$bhp_{rated} = P_0 + P_1 \cdot T_{cond} + P_2 \cdot T_{suc} + P_{11} \cdot T_{cond}^2 + P_{22} \cdot T_{suc}^2 + P_{12} \cdot T_{cond} \cdot T_{suc} \quad (12)$$

The compressors in this refrigeration system, like many refrigeration systems, operate in a base/trim manner, which means that the last compressor to turn-on for each stage is the trim compressor. Equation 4, which is analogous to Equation 12, shows the full-load capacity of the compressor as a function of the suction and condensing temperatures under-which the compressor operates. Knowing the refrigeration load (Ref Load) and capacity of the base compressors that are operating ($\sum TR_{Base}$), the fraction capacity of the trim compressor for a given stage (FC_{Trim}) can be calculated as:

$$FC_{Trim} = (\text{Ref Load} - \sum TR_{Base}) \;/\; TR_{Trim} \quad (13)$$

If the capacity of the base compressor is greater than or equal to the refrigeration load, then refrigeration provided by the base compressors is zero.

Typically, the part-load power of a compressor varies based on fraction capacity and the pressure ratio. It is best to determine the part-load power of each compressor using field measurements. For the part-load power of the compressors in this analysis, a typical part-load performance curve was used (Stoecker 1998). The fraction-power of the compressor is of the following form:

$$FP = a_0 + a_1 \cdot FC + a_2 \cdot FC^2 + a_3 \cdot FC^3 + a_4 \cdot Pr \quad (14)$$

For a trim compressor, the brake horse-power at part-load can be modeled as:

$$bhp_{act} = bhp_{rated,Trim} \cdot FP \quad (15)$$

Using the rated motor efficiency and calculated shaft power of each compressor, total input power of all compressors ($\sum P_{Compressors}$), in units of kilowatts, can be calculated as:

$$\sum P_{Compressors} = \sum bhp_{act} \;/\; \eta_m \cdot 0.746 \text{ (kW/hp)} \quad (16)$$

Calculating Condensing Pressure

The capacity of an evaporative condenser increases as the condensing temperature increases and the ambient wet-bulb temperature decreases. In addition, power draw of the compressor increases with increasing condensing temperature and refrigeration load. Energy use of both condensers and compressors changes significantly with condensing temperature; accurately determining the actual operating condensing temperature is essential to quantify energy use of the system.

Figure 5 shows the condenser capacity for high and low wet-bulb temperatures and the heat rejection from the compressors for high and low refrigeration loads over a range of condensing temperatures. The rectangles show where the condenser capacity and compressor heat rejection are equal. Notice that for low wet-bulb temperatures and low refrigeration loads, the condenser has enough capacity to reject compressor heat at even lower condensing pressures. However, in practice most refrigeration systems are not able to operate at condensing pressures of 90 psig or less due to system constraints. Therefore, a condensing pressure set-point maintains a minimum condensing pressure by controlling the condenser fans and pumps. Typically at low wet-bulb temperatures and low refrigeration loads, the system operates at the minimum condensing pressure set-point.

Figure 5. Condenser Capacity and Compressor Heat Rejection with Different Set-Point Pressures.

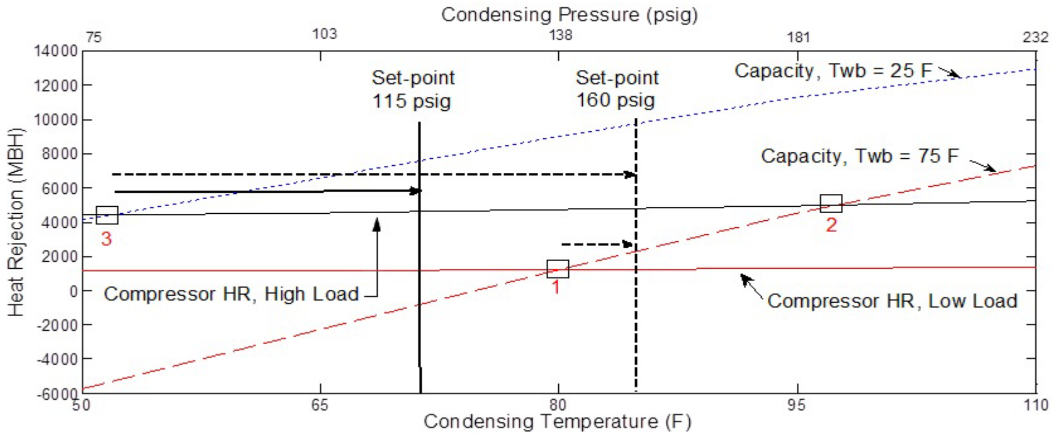


Figure 5 also illustrates the operating condensing pressures for condensing pressure set-points of 115 psig and 160 psig. For a set-point pressure of 160 psig at low refrigeration loads and high wet-bulb temperatures, state 1, the fans and pumps will be controlled such that the system operates at 160 psig. However, if the condensing pressure set-point is reduced to 115 psig, the fans and pumps will operate at 100% capacity and the condensing pressure will be about 138 psig. At high wet-bulb temperatures and refrigeration loads, state 2, the condensing pressure will be about 190 psig and the fans and pumps will operate at 100%, regardless of the set-point pressure selected.

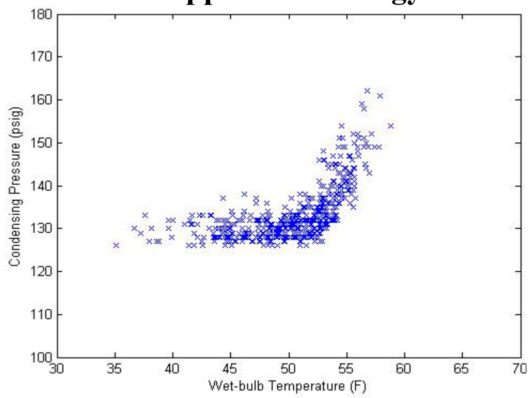
At state 1, as the set-point pressure is reduced from 160 psig to 115 psig, the condensing pressure will reduce from 160 psig to about 138 psig, which will reduce compressor energy use. However, this will cause the condenser fans to operate at full speed, increasing energy use of the condenser fan.

At low refrigeration loads, compressor energy is significantly less than at high refrigeration loads and small reductions in compressor energy use can be off-set by large increases in condenser fan energy use. This situation is common in the food processing industry where production occurs on weekdays and freezers and coolers are in holding mode on weekends.

To reduce energy use during periods of high wet-bulb temperatures, a wet-bulb approach strategy can re-set condensing pressure set-point based on the temperature difference between condensing temperature and wet-bulb temperature. For example, at a wet-bulb temperature of 75^{o}F and an approach temperature of 10^{o}F, the condenser fans will be controlled to maintain a condensing pressure set-point of 85^{o}F (151 psig). Figure shows measured condensing pressure versus wet-bulb temperature data from the system under study. The system uses a wet-bulb approach strategy with a wet-bulb approach temperature of 14^{o}F and a minimum condensing temperature and pressure of 70^{o}F and 130 psig respectively. The sloped part of the data show

how condensing pressure increases with increasing wet-bulb temperature. The flat part of the data is caused by the minimum condensing pressure set-point, which cycles condenser fans to maintain the minimum pressure set-point. The scatter is caused by refrigeration loads which vary even at the same wet-bulb temperature and by dynamic effects within the system.

Figure 6. Condensing Pressure versus Web-bulb Temperature with Ambient Wet-bulb Approach Strategy.



Knowledge of the minimum condensing pressure set point facilitates calculation of condensing pressure. To do so, Equation 17 can be used to calculate the heat rejection of the compressors and Equation 18 can be used to calculate condenser capacity. If the condenser capacity meets or exceeds the heat rejection from the compressors at set-point pressure for a given wet-bulb temperature and refrigeration load, then the system operates at the set-point pressure. But if heat rejection from the compressors exceeds the condenser capacity, then the system operates at condensing pressure higher than the set-point pressure.

When compressor heat rejection exceeds condenser capacity, the heat of compression for the compressors, refrigeration capacity of the compressors and heat rejection capacity of the condensers should first be evaluated at the highest condensing temperature to determine the brake horse-power (Equation 12), rated tons of refrigeration for the compressors (Equation 4) and heat rejection capacity of the condenser (Equation 9a,10,11). This is referred to as the high condensing temperature limit for simulation purposes. If the condenser capacity is still insufficient, then the condensing temperature is out of range and further action is required to resolve this issue.

If the condenser capacity exceeds the heat rejection of the compressors at the high condensing temperature, then condensing pressure can be determined by first curve-fitting both compressor heat rejection and condenser capacity over a range of condensing pressures from set-point pressure to the high condensing temperature limit. Inspections of the heat rejection curves in Figure 5 suggest that these curves should be fitted to a second order polynomial. Thus, the heat rejection from the compressors for a given wet-bulb temperature can be curve fitted as:

$$\text{HR Compressors} = k_0 + k_1 \cdot T_{cond} + k_2 \cdot T_{cond}^2 \quad (17)$$

And the capacity of the condensers for a given wet-bulb temperature can be curve-fitted according to the following equation:

$$\text{Condenser Capacity} = m_0 + m_1 \cdot T_{cond} + m_2 \cdot T_{cond}^2 \quad (18)$$

The condensing temperature is the temperature at which heat rejection from compressors and condenser capacity are equal within the range of set-point pressure to high condensing pressure limit. Hence the condensing temperature can be solved for using Equations 17 and 18.

Calculating Condenser Fan Power

If the system operates at a condensing pressure above set-point, the condenser fan will continue to operate at full speed. However, when the condenser has enough capacity to maintain the set-point at a given wet-bulb temperature and refrigeration load, the fans will be controlled to maintain set-point pressure. The fraction capacity of the condenser is then:

$$FC_{Condenser} = \text{HR Compressors} / \text{Condenser Capacity} \tag{19}$$

The fraction capacity of the condenser can be met by either cycling the fans on and off or reducing fan speed. For on/off control, the fraction on time (F_{on}) is the same as $FC_{Condenser}$ and so fan power can be calculated using fraction on and the input power of the fan $P_{fan,full}$ as:

$$\text{Fan Power} = F_{on} \cdot P_{fan,full} \tag{20}$$

When fan speed is reduced, percent flow (% Flow) can be calculated using a heat exchanger exponent, HX_{exp}, as (Mitchell & Braun 1998):

$$\%\text{ Flow} = (FC_{Condenser} / \text{constant})\ \hat{}\ (1/HX_{exp}) \tag{21}$$

This analysis assumes a constant of 1.00 and HX_{exp} of 0.80.

Typically, VFDs are used to reduce fan speeds. Fan power of the condenser fans controlled with VFDs closely follows the fan affinity laws with an exponent of 2.7. Assuming constant VFD losses of 3%, fan power for condenser fans can be modeled as:

$$\%\text{ Power} = 1.03 \cdot P_{fan,full} \cdot (\%\text{ Flow})^{2.7} \tag{22}$$

Annual Simulations and Economics

ECMs for Retrofit Applications

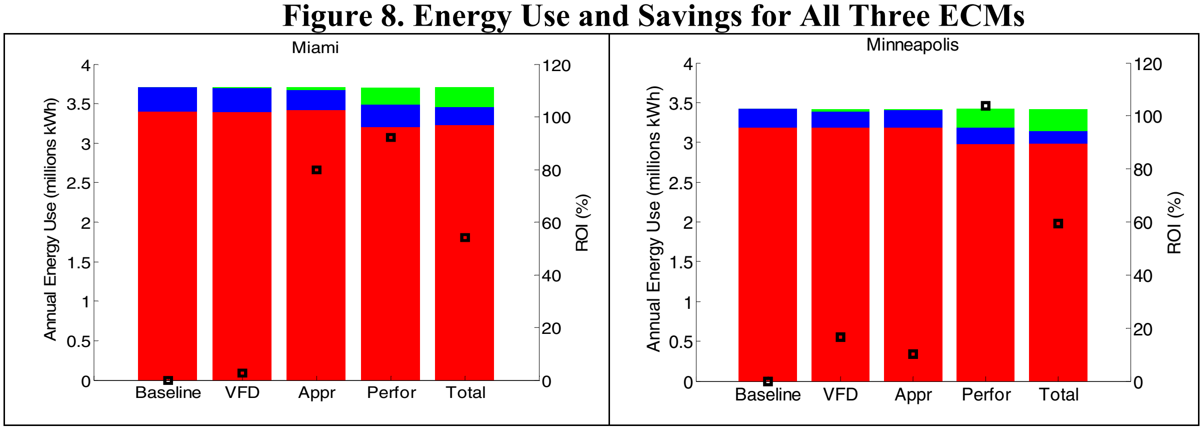
Annual simulations using this methodology and TMY3 (NREL 2005) weather data were performed for three energy conservation measures (ECMs): installing a VFD on condenser fan, using wet-bulb approach strategy and increasing condenser performance. The following bi-modal load profile was used; during weekday operations between 7 am and 11 pm, the system operates with a constant low temperature refrigeration load of 175 tons and a high temperature load of 100 tons. The refrigeration load is 25% of these values for all other times.

The cost effectiveness of each measure was evaluated by calculating the simple return on investment (ROI). The costs of these three measures are typically modest. The cost to install a 50 hp VFD is about $13,000. The cost for control modifications for a wet-bulb approach strategy is about $3,000. The cost of fixing belts and fans and removing scale is about $15,000. The 2012 average cost of industrial electricity of $0.067/kWh (EIA, 2013) and annual energy escalation rate of 3% is used to calculate cost savings over the 20-year lifetime of each measure.

Figure 8 shows the energy savings and return on investment (ROI) for each ECM and the total of all three ECMs if the plant were located in Miami, FL and Minneapolis, MN. By far the biggest energy saving opportunity for this existing system with an under-performing condenser is to increase the condenser performance. This can be achieved by fixing belts, fans, nozzles and removing scale. It is also of note that the energy savings for installing all three ECMs in these cases is higher than the sum of the savings for each individual ECM.

In both locations, improving condenser performance is the most cost effective energy conservation measure. The cost effectiveness of adding a VFD on the condenser fan and utilizing wet-bulb approach controls depends upon the ambient weather conditions of the region. For example, in Miami, the hot and humid weather does not allow the system to operate at the minimum condensing pressure set point, and hence installing a VFD on the condenser fan rarely results in energy savings. However, employing the wet-bulb approach strategy with a VFD on the condenser fan allows the condensing pressure set-point to be varied to achievable levels. This results in higher energy savings and an attractive ROI of 80%. In cold regions like Minneapolis, MN the minimum condensing pressure set-point can frequently be met and adding a VFD on the condenser fan has a better ROI than utilizing a wet-bulb approach strategy. These results demonstrate the complexity of determining the cost effectiveness of energy efficiency measures for ammonia refrigeration systems. They underscore the need for accurate modeling, such as the modeling presented here, to make good decisions.

Figure 8. Energy Use and Savings for All Three ECMs



Red = Compressor Energy, Blue = Condenser Fan Energy, Green = Energy Savings, Square = ROI, VFD = constant to variable speed condenser fan, Appr = utilizing wet-bulb approach strategy and Perfor = improving condenser performance.

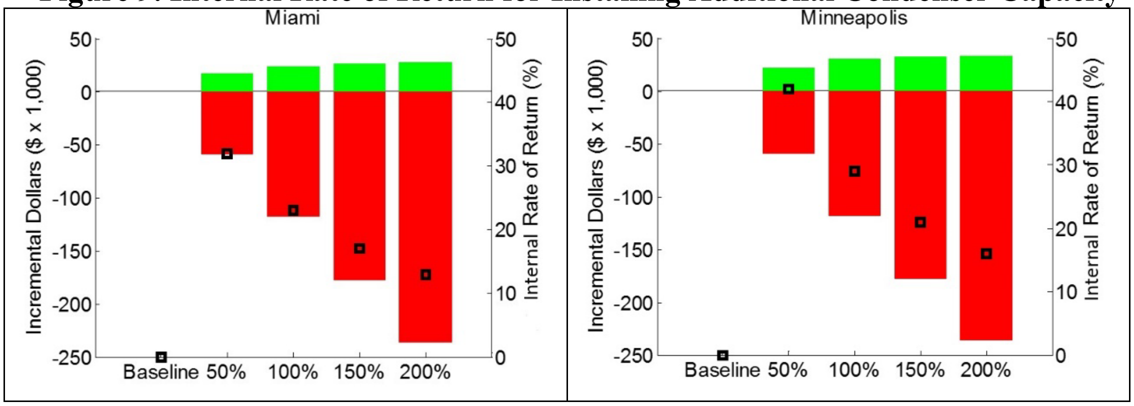
Installing Additional Condenser Capacity in New Construction Applications

Condensers are costly to install because of the structural supports, piping and controls. Therefore it is seldom cost-effective to install additional condensers for the sole purpose of energy efficiency. However, in new construction installing additional condenser capacity can be cost effective. The approximate installed cost for a condenser with VFDs and wet-bulb approach controls is represented in Equation 23.

$$\text{Incremental Cost (\$)} = 17 \cdot \text{Incremental Capacity (MBH)} + 12{,}000 \qquad (23)$$

Figure 9 shows that the internal rate of return (IRR) when adding additional condenser capacity when the life of a condenser is 20 years and the energy escalation rate is 3%. The IRR was calculated for installing additional capacity of 50%, 100%, 150% and 200% over the baseline capacity of 7,000 MBH. In both locations, the internal rate of return exceeds 20% for doubling condenser capacity. Thus, increasing condenser capacity appears to be a very attractive option for new construction.

Figure 9. Internal Rate of Return for Installing Additional Condenser Capacity



Red = Incremental Cost, Green = Annual Energy Cost Savings, Square = Internal Rate of Return

Summary and Discussion

This paper developed a methodology to calibrate condenser performance using data from a refrigeration control system. This calibrated condenser performance was used in a simulation model to calculate the annual energy use of the system under study. The simulation model was then used to calculate energy savings for three ECMs: installing a VFD on the condenser fan, using wet-bulb approach strategy and increasing condenser performance for two different ASHRAE climate zones.

Important results are:

1. Total power consumption of refrigeration systems is strongly dependent on condenser size, performance and control.
2. For existing systems, improving evaporative condenser performance may be the most cost effective energy conservation measure. Visual inspection for scale and damage may identify some degradation; however, determining the actual effectiveness as shown in Figure 3 requires that the system be modeled using the method developed here.
3. Energy savings from adding VFDs on condenser fans and utilizing wet-bulb approach control is dependent on the ambient weather conditions of the region. In colder climates it is better to install VFDs on condenser fans. However in hot and humid climates it is better to install VFDs on condenser fans AND utilize a wet-bulb approach strategy.
4. Doubling the condenser capacity during the design phase can result in an internal rate of return above 20% and allows for future system expansion.
5. If multiple opportunities are implemented simultaneously, the resulting savings are typically greater than the sum of individual savings.

In summary, the paper presents an integrated approach for design, retrofit and control of evaporative condensers in ammonia refrigeration systems. The methodology presented here can assess the feasibility of energy efficiency measures for both existing and new systems. Energy savings derived through the use of this approach can improve energy efficiency by about 10% or more even in diverse climates.

Future work will include application of this method to ammonia refrigeration systems with multiple condensers. In multiple-condenser systems, condenser pump power is not negligible. It should be considered along with other control strategies such as condenser staging and analysis of dry condenser operation in order to optimize energy efficiency.

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