High Efficiency Evaporative Condensers for Air Conditioning, T.E.S., and Refrigeration

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Evaporative condensers are a proven, cost-effective method of increasing air conditioning and refrigeration capacity, and reducing peak demand. They provide the most energy efficient method for rejecting heat from any vapor compression system—increasing system efficiency by more than 20%. However, they are typically used for systems over 50 tons of capacity. Use of evaporative condenser technology increases efficiency in commercial and industrial refrigeration, thermal energy storage (TES), and commercial air conditioning.

Evaporative condensers are installed in cooling and refrigeration systems to reject heat and may be used with all types of compressors with high pressure refrigerants. To optimize compressor energy efficiency, which is defined as fan plus pump horsepower per quantity of heat rejection, evaporative condensers are designed based on an approach temperature of 80 F to 12°F; rather than a specific condensing temperature.

For optimum condenser efficiency, the amount of surface in the condenser should be maximized and the fan-and pump-horsepower minimized. For operation at low loads, two-speed energy efficient motors should be used rather than adjustable speed drives.

Whether adding evaporative condensers to an existing system, or designing new systems, they must be properly piped, installed, and maintained. From an energy efficiency standpoint, the majority of evaporative condensers are improperly installed. This paper provides detailed steps to calculate the quantity and cost of saved energy, as well as suggestions to maximize evaporative condenser efficiency.

Applications

As mentioned above, energy efficient evaporative condensers are used in air conditioning, thermal energy storage (TES), and commercial and industrial refrigeration systems. Grocery and convenience stores can also take advantage of improved heat rejection equipment. This is especially true in the northern climates, where the use of evaporative condensers is expanding, as freeze problems have been overcome.

Water source heat pumps and chiller systems can also achieve similar savings with improved heat rejection equipment. Improved efficiency evaporative condensers can reduce chiller peak demand by up to 15%. Thermal energy storage systems use high efficiency evaporative condensers. Doubling or tripling the amount of condenser surface compared to typical design is cost effective in many climate zones.

Table 1. Impacts on Performance

Item	Range of Degradation, %
Placement	10 to 20%
Piping	10 to 50%
Purging	10 to 30%
Fouling	10 to 30%
Wet-bulb	10 to 20%

Increasing the capacity of the heat rejection equipment versus typical design practice and limiting compressor capacity based on the heat rejection capacity will result in at least a five percent energy saving compared to a standard chiller/TES system. For refrigeration/TES systems, a ten percent energy saving is possible. Performance will vary with climate zone, compressor efficiency, and suction temperature.

Evaporative condensers are used with centrifugal, reciprocating, scroll, and screw compressors over 50 tons of capacity. They are used to desuperheat booster discharge gas in two-stage grocery store and industrial refrigeration systems. Desuperheating the discharge gas, as shown in Figure 1, will improve high stage refrigeration efficiency by about four percent. In addition, water consumption in the evaporative condenser is slightly reduced.

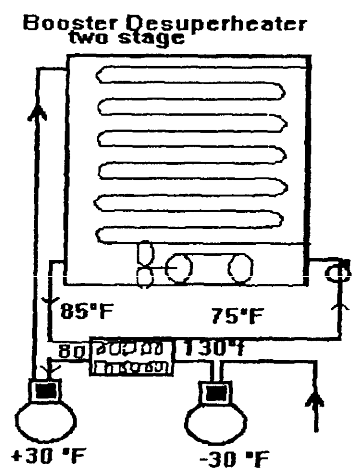


Figure 1. Booster Desuperheater Two Stage

Target Market

Almost all vapor compression equipment over 50 tons can use evaporative condensers. Table 2 shows a sample of the target markets, and potential demand reduction with energy efficient, close approach condensers. [1]

Refrigerants

Evaporative condensers are used with all high pressure refrigerants, such as HCFC-22, HFC-134a, and ammonia. (Evaporative condensers cannot be used with CFC-11 nor with HFC-123 refrigerants, due to the vapor density). A specified evaporative condenser using ammonia refrigerant is about 7% more efficient than if it is using HCFC-22 or HFC-134a refrigerant. Evaporative condensers can be used for HFC-134a refrigerant using manufacturers' data for HCFC-22 refrigerant; however the refrigerant pressure drop will be slightly higher. Table 3 displays the typical condensing temperatures associated with a less-efficient design.

Table 2. Target Markets and Potential Demand Reduction

Market Potential kw/ton	Demand Reduction
Supermarket	
Bakery/Meat rack	0.28
Product/Dairy rack	0.2
Refrigerated Warehouse	
Coolers (+20°F)	0.12
Freezers (-10°F)	0.2
Food Processing	
Hydrovacs/Vacuum Tubes	0.1
Ice Production (+10°F)	0.15
IQF/Plate Freezers	0.18
HVAC	
Air Conditioning	0.14
TES	0.2

Assumptions: Peak condensing temperature decrease from 105°F to 85°F for HCFC-22 refrigerant systems; and from 95°F to 85°F for ammonia refrigerant systems. Actual demand reduction will vary by system.

System Capacity Versus Condensing Temperature

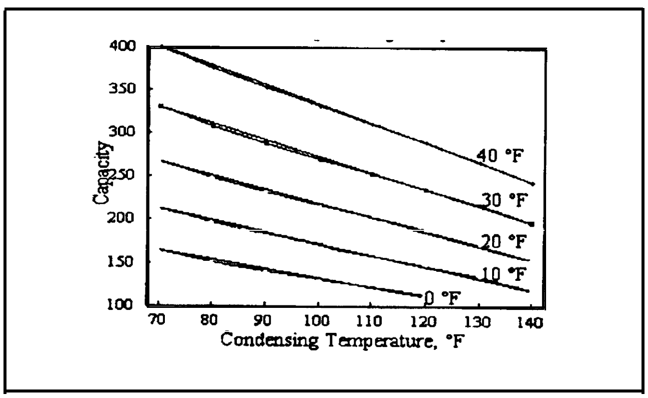
Just as the system efficiency increases as the condensing temperature decreases, the system capacity also increases with decreasing condensing temperature. It is more energy efficient and cost-effective to add more condenser surface when additional capacity is needed in an air-conditioning or refrigeration system. Graph 1 shows system capacity versus condensing temperature for different suction temperatures.

These temperatures should not be used for efficient design. Previous applications have indicated that an approach temperature of 8-12°F will minimize the condensing temperature to maximize capacity and energy efficiency.

Table 3. Typical Inefficient Design Condensing Temperatures

Refrigeration	
Ammonia	95°F
HCFC-22, HFC-134 and HVAC	105°F
Chiller, cooling tower	100°F
Unitary, air cooled	125°F

These temperatures should not be used for efficient design. Instead, use an approach temperature of 8-12°F. Minimize the condensing temperature to maximize capacity and energy efficiency.



Graph 1. Capacity vs. Condensing Temperature HCFC-22 Reciprocating Compressor

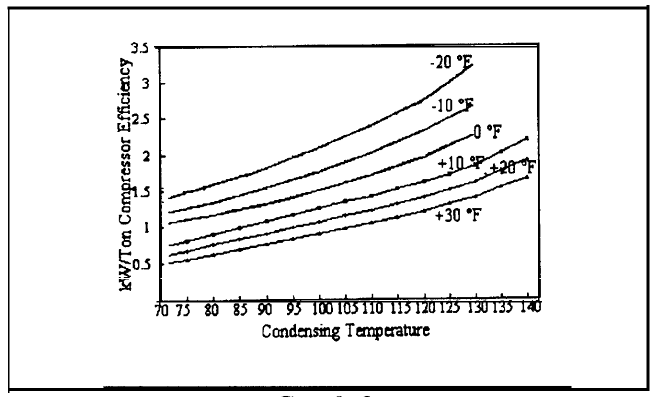
Efficiency

Compressor energy usage and capacity are a function of the suction and condensing temperature. This is true for all compressors, chillers, and refrigerants.

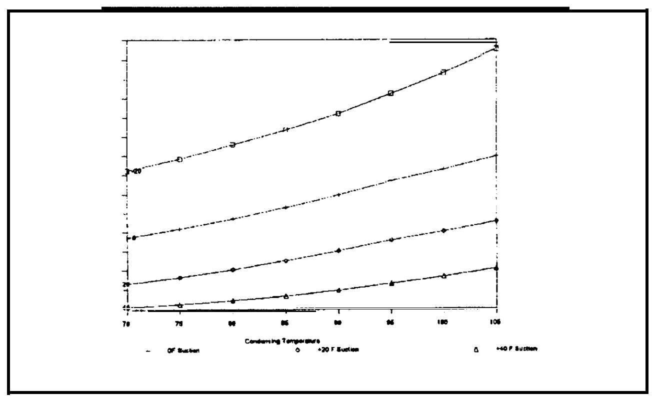
The following items impact condensing temperature:

- ambient wet-bulb temperature,
- available condenser heat exchange surface and scaling,
- compressor efficiency,
- refrigerant,
- interaction of the heat exchange surface with the air and refrigerant,
- air and water distribution,
- compressor load,
- expansion valve setting.

Energy usage increases and capacity decreases as the condensing temperature increases. As graphs 2 and 3 show; refrigeration system kw/ton reduction is not linear with the reduction in condensing temperature. An HVAC unit with an air cooled condenser sitting on a hot roof will use far more energy than one with a cooling tower.



Graph 2.



Graph 3. Ammonia Compressor Efficiency vs. Discharge Temperature, Suction Temp.

Condenser Efficiency

There is over a 500% variation in efficiency between the most and least efficient units. Condenser efficiency is measured in fan plus pump horsepower per ton. Perusal of major manufacturer's catalogs reveals that the most efficient practical unit runs about 0.03 horsepower per corrected evaporator ton; and that the least efficient unit runs about 0.17 horsepower per corrected evaporator ton. That is, per 1,000 tons of air conditioner or refrigeration load, the condensers total fan plus pump horsepower might have a difference of 140 horsepower.

Manufacturers have responded to first cost pressures by reducing the amount of surface per ton, and by increasing fan horsepower. As discussed above, the three major components of condenser capacity are surface area, air movement, and water distribution. According to major manufacturers, it is less costly to install a 30 hp fan motor and reduce surface area, than to use a 20 hp fan motor. To maximize energy efficiency, add heat exchange surface and reduce fan horsepower.

Reducing fan horsepower by 40% results in a 12% reduction in condenser capacity. Note that this relationship varies with different manufacturers. Often fan motor horsepower can be reduced by two sizes (such as going from 50 hp to 30 hp) if another row of coils is added to compensate for the capacity change.

Figure 2 shows the relationship between fan plus pump horsepower, coil surface, and condenser capacity. Reducing the fan plus pump horsepower from 55 hp to 36 hp and adding 4 more rows of condenser surface increases capacity from approximately 950 tons to 1140 tons. The incremental cost of adding surface area may payback in less than two years, depending on utility incentives, rates, and operating hours.

Axial and Centrifugal Fans

Axial fans (also known as propeller fans) typically use less energy than centrifugal fans. Condensers with centrifugal fans use roughly 30 to 40% more energy (fan and pump horsepower per ton) depending on model and ton capacity. However, axial fans are typically far noisier than centrifugal fans, and have therefore had poor acceptance in air conditioning and supermarket applications.

Some manufacturers are supplying lower speed wide chord axial fans that reduce the noise level to slightly more than that of centrifugal fans.

Motors and Adjustable Speed Drives

Single speed energy efficient fan motors are used in evaporative condensers. To optimize efficiency during periods of low load, low ambient temperature conditions, use two-speed energy efficient motors or pony motors. Use caution in specifying two-speed or pony motors. It may be more efficient for the system to operate the fans at full speed and obtain lower condensing temperature, than to operate the motors at partial load and have a higher condensing temperature.

Adjustable speed drives (also known as inverters) have been used with varying success, and are not typically cost-effective. Analysis by manufacturers indicates that adjustable speed drives rarely save energy as increased compressor energy usage offsets fan energy savings.

Proper determination of the temperature at which to control fans will depend on the compressor type (screw versus reciprocating or scroll) and efficiency at part load and different condensing temperatures, the suction temperature, and condenser efficiency in fan horsepower per ton. This is especially critical with screw compressors operating at partial load, and with different Vi ratios.

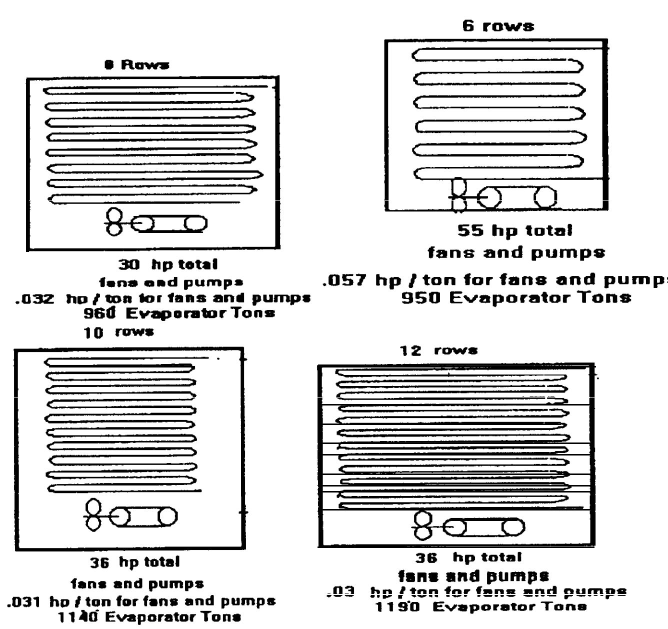
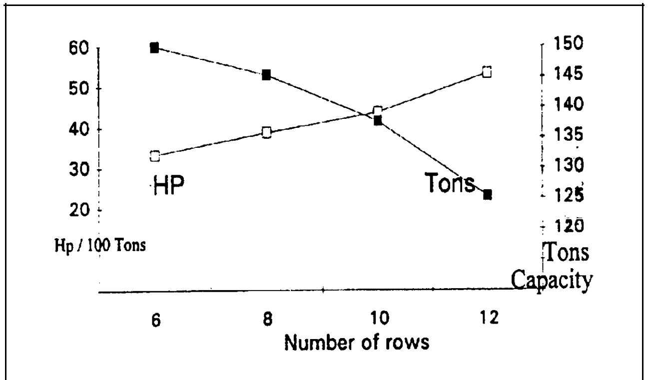


Figure 2. Relationship of Fan and Pump Horsepower vs. Capacity

Condenser Capacity

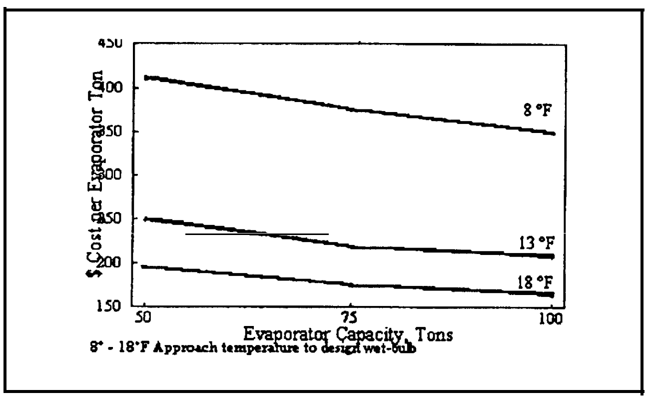
Heat rejection capacity per unit surface area decreases as the number of rows increases. Also, the fan horsepower per ton increases. Fan horsepower per ton increases due to increased static pressure drop that the fan must overcome as the number of rows increases. Heat rejection per row (per unit surface area) decreases due to decreased temperature differentials in the condenser. Graph 4 shows this relationship. However it may be possible to reduce the fan horsepower per ton and increase the number of rows and still increase heat rejection capacity.



Graph 4. Typical Fan Horsepower Pr 100 Evaporator Tons, and Condenser Capacity, vs. Row Depth

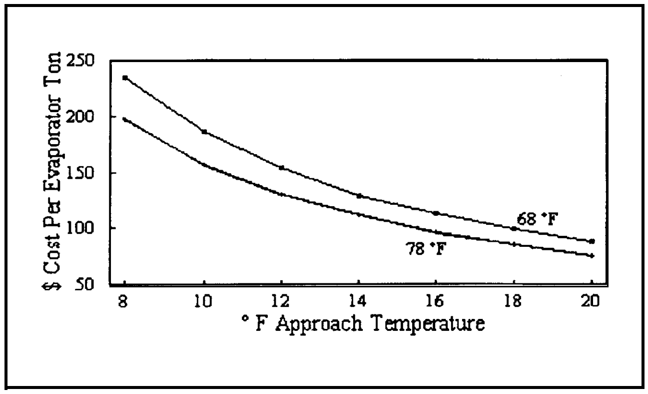
Condenser Cost

The cost of evaporative condensers will vary with the approach temperature and design wet-bulb. Cost will also vary by geographical area and by contractor. Graph 5 shows average cost per evaporator ton for an HCFC-22 TES or supermarket refrigeration system (20°F suction temperature) versus approach temperature, for 72°F design wet-bulb temperature.[2]



Graph 5.

Graph 6 shows average cost per evaporator ton for an ammonia system, for 68°F and 78°F wet-bulb. As can be seen from the above charts, the incremental cost per ton is lower for large ammonia evaporative condensers than for smaller HCFC-22 or HFC-134a evaporative condensers. This is due to the economies of scale in manufacturing. To offset these, utilities may consider offering larger rebates per ton for small tonnage condensers.



Graph 6. Large Ammonia Refrigerant Condenser Cost Per Ton vs. Approach Temperature (68°F and 78°F Wet-Bulb)

Impacts on Performance

The majority of condensers are improperly installed. Changing the piping or placement of the condensers has resulted in large energy savings. Likewise, large energy penalties have been observed with improper selection of wet-bulb temperature and poor maintenance. Table 1 lists the range of performance degradation of the following items.

Placement

Condensers need to have plenty of room to prevent recirculation from the exhaust stack back to the air inlet. The air inlet side of a condenser should be at least 5 feet away from the nearest wall for a 100 ton unit, and up to 12 feet away for a 1500 ton unit. The remaining sides of the condenser should be at least four feet from the nearest wall or structure. Double these figures if multiple condensers are installed. The exhaust stack of the condenser should be higher than surrounding walls or structures. Condensers should be located to prevent the introduction of discharge air into building ventilation systems. The prevailing wind should enter or be perpendicular to the air inlet. Consult manufacturer for detailed information.

Piping

Improper piping is a major problem that is seldom detected. For optimum low temperature performance ("floating the head pressure"), the drop leg should be a minimum of

8-10 feet in height. There should be a minimal pressure drop across the condenser and liquid drop leg. The condenser plus piping pressure drop must be identical if multiple condensers are used. The condenser outlet should be piped to the bottom of the receiver, so that sub cooled liquid exiting the condenser flows directly to the expansion device. Installation of an equalizing pipe between the top of the receiver and the inlet of the condenser will ensure that the condenser drains properly.

Valves on liquid drop leg must have a low pressure drop, and be located near the bottom third of the leg. Valves should not be installed on the horizontal lines. Liquid drop leg pipes should be sized for a velocity of no more than 100 fpm. Size gas inlet lines for a maximum of 40 fps.

Floating Head Pressure

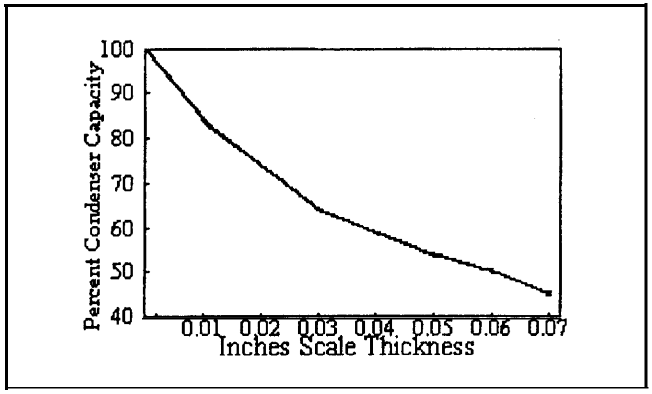
Floating head pressure is paired with piping because most systems are improperly piped to allow for this energy efficiency measure. Floating the head pressure, or condensing pressure, refers to reducing the condensing pressure as low as possible when the ambient wet-bulb temperature decreases. Some energy efficient systems float the head pressure as low as 40°F, significantly reducing compressor energy consumption with a modest increase in condenser fan energy consumption. Typically, condensing temperature is controlled when it reaches 85°F or, when the ambient temperature reaches 60°F, by cycling condenser fans to half speed or turning them off. Maintaining a high condensing pressure is done when a large pressure differential is needed to operate older model expansion valves, or to account for inappropriate piping practices. Piping, valving, or other modifications may be needed to ensure that the system will operate properly at lower condensing pressures.

Purging

Air and other vapors that do not condense at the refrigerant condensing temperature are called non-condensables. Non-condensables migrate to the highest point of a system, and degrade system performance by reducing the amount of available heat transfer surface, Non-condensables are in almost all systems, including systems operating above atmospheric pressure, such as HCFC-22 and HFC-134a air-conditioning systems. Consider an automatic purger to remove non-condensables in systems that are opened periodically, and in systems with pumps or gaskets. Install purge valves at the high point of the chiller bundle, the high point of the system, the receiver, and at each condensing coil outlet.

Fouling

A small increase in exterior scale buildup will result in a significant decrease in performance. It is necessary to keep condensers clean, or their performance will degrade. Emphasize a proper maintenance program as part of the energy efficiency recommendations. Some utilities providing incentives for energy efficient condensers require a 5 year manufacturer approved maintenance program.



Graph 7.

Proper Wet Bulb

A 20 F increase in wet-bulb temperature decreases condenser performance by 5-15%, depending on peak-design condition. Therefore it is critical that the proper wet-bulb temperature be used. Recommend that ASHRAE 0.5% wet-bulb temperature be used, In some cases it may be necessary to add a few degrees to account for local rivers, lakes, process exhaust, or recirculation of condenser discharge air.

Calculating Energy Savings

Demand reduction of several hundred kW; and energy savings close to a million kWh; have been documented in individual facilities with improved heat rejection equipment. The largest potential for demand reduction per end-use customer is with refrigerated warehouses and food processors.

Presented below are two methods of calculating energy savings. The first method offers a first order approximation of peak demand reduction per ton; the second method affords a more rigorous analysis of annual energy savings. Use either method to determine energy savings by reducing fouling, reducing non-condensables, adding additional heat exchange surface, floating head pressure during periods of low ambient condition, or reducing the local wet-bulb temperature with the installation of exhaust hoods or decking.

The first method starts by estimating the reduction in peak condensing temperature.

Tables 4[3] and 5[4] show typical kw/ton demand reduction for every 10°F delta T reduction in condensing temperature, for different condensing and saturated suction temperatures. To determine the demand reduction, multiply the peak evaporator tons by the kw/ton/10°F delta T factor, and the ratio of condensing temperature reduction and 10°F. Note that the number of peak tons may be different from the compressor capacity. Estimate Annual kWh energy reduction by multiplying the peak kW demand reduction by the equivalent full-load hours.

Table 4. Typical kw/ton/10°F Delta T Factors-HCFC-22

Condensing	Suction Temperature			
Temperature	-20	0	20	40
120°F	0.34	0.25	0.17	0.14
105	0.28	0.20	0.16	0.14
100	0.27	0.18	0.16	0.13
95	0.26	0.17	0.15	0.12

Table 5. Typical kw/ton/10°F Delta T Factors-Ammonia

Condensing	Suction Temperature			
Temperature	-20	0	20	40
95°F	0.19	0.14	0.11	0.06
90	0.16	0.12	0.10	0.05
85	0.15	0.11	0.09	0.04
80	0.14	0.10	0.07	0.03

Tables 4 and 5 show typical kw demand savings by reducing the condensing temperature 10°F for HCFC-22 and ammonia refrigerants (Example: go from 120 to 110°F for a 10°F delta T); for various suction temperatures.

Example: Currently a 100 ton HCFC-22 supermarket refrigeration system operating at +20°F suction temperature has a peak condensing temperature of 100°F, A contractor states that the tubes have roughly 0.01" thickness of scale build-up; and that there is roughly a 2°F increase in local wet-bulb temperature due to exhaust air recirculation. Design wet-bulb temperature is 68°F. The contractor offers to clean the condenser tubes and install an exhaust hood to prevent recirculation. Assume 3,000 equivalent full load hours.

Step 1. Assume the tubes will be cleaned first. Note that the local wet-bulb temperature is 68 + 2 = 70°F. From Graph 7, "Scale Thickness versus Capacity, " notice that the condenser capacity is roughly 85% of the original for 0.01" scale buildup. From Table 8, "HCFC-22 Net Refrigeration Effect Correction Factor (CF-l)," note that the application factor for 70°F wet-bulb and 100°F condensing temperature is 1.03 and that the application factor for 95°F condensing temperature is 0.84. To determine the application factor when the tubes are clean, multiply the 1.03 application factor by 0.85 = 0.88. By interpolation from Table 8, the new condensing temperature will be approximately 96°F.

Step 2. From Table 8, notice that at 68°F wet-bulb and 95°F condensing temperature the application factor is O. 89; and that at 90°F condensing temperature the application factor is 0.70. From step 1 the application factor is 0.88 with clean tubes. The new condensing temperature will be slightly less than 95°F.

Step 3. Determine the delta T in condensing temperature. Subtract the new condensing temperature, 95°F, from the original condensing temperature, 100°F, to determine a 5°F delta T.

Step 4. From Table 4 "Typical kw/ton/l0°F delta T Factors for HCFC-22," note that the factor is 0.16 kw/ton/10°F delta T for 20°F suction and 100°F condensing temperature. The demand reduction will be 0.16 kw/ton of evaporator capacity by decreasing the condensing temperature 10°F. Multiply the 100 ton evaporator load by 0.16 kw/ton, times 5°F/l0°F, = 8 kW demand reduction.

Step 5. To estimate the annual energy savings, multiply the kW demand reduction by the equivalent full load hours: kWh savings = 8 kW x 3,000 hours = 24,000 kWh.

Bin Temperature Method

The second method of estimating energy savings is with a bin temperature analysis. A bin temperature analysis divides the year into 5°F increments, and has the number of hours at each increment, and the corresponding mean coincident wet-bulb temperature. This method accounts for variation in load and wet-bulb temperature throughout the year.

To use this method, begin with the application factor for the condenser. For each temperature bin:

- ratio the application factor by the load in tons versus the peak load
- ratio the application factor by the wet-bulb correction factor from Table 8 "Net Refrigeration Effect Correction Factor (CF1)" versus the peak wet-bulb temperature
- Determine the condensing temperature based on the mean coincident wet-bulb temperature and the corrected application factors.
- For that condensing temperature and load, determine the compressor kW from manufacturers data, accounting for part loading penalties.
- Multiply the compressor kW plus the condenser fan and pump kW by the hours in temperature bin to determine kWh.

To determine total kWh sum the kWh in each temperature bin (See Tables 6 and 7).

Placement Calculations

Calculate energy savings by estimating the existing wet-bulb at the air inlet, and the new wet-bulb after an exhaust hood or decking is installed to reduce air recirculation. Then use Table 8 or 9 (depending on the refrigerant) to determine the change in condenser capacity; and the subsequent new condensing temperature. From Table 4 or 5 (depending on the refrigerant) determine the change in compressor kW.

Piping Calculations

Estimate energy savings by determining the existing condensing temperature and the condensing temperature after modifying the piping. Determine the change in compressor kW from either Table 4 or 5.

Table 6. Standard Approach Evaporative Condenser Bin Hour Analysis of kW and kWh (HCFC-22 Reciprocating Compressor, Evaporative Condenser Sized for a 1.2 Total Application Factor)

Bin Temp.	Hour Year	Wet-Bulb	Load Tons	Cond. Temp.	Comp. kW	Fan kW	Total kW	kWh
105/109	135	68	100	105	115	4.7	119	16104
100/104	315	66	100	102	110	4.7	115	36094
95/99	406	65	100	102	110	4.7	115	46521
90/94	508	63	100	101	108	4.7	113	57411
85/89	593	60	100	99	105	4.7	110	65156
80/84	636	58	100	98	104	4.7	108	68883
75/79	638	55	100	96	101	4.7	105	67097
70/74	645	52	100	95	99	4.7	104	66821
65/69	678	49	100	94	97	4.7	102	69176
d60/64	724	47	100	104	113	1.5	115	82969
55/59	781	44	75	101	81	1.5	83	64668
50/54	765	41	75	98	78	1.5	79	60642
45/49	669	38	75	95	74	1.5	76	50670
40/44	514	35	75	93	72	1.5	73	37720
35/39	361	31	75	90	68	1.5	70	25218
30/34	234	27	75	90	68	1.5	70	16346
25/29	94	23	75	90	68	1.5	70	6566
20/24	25	18	75	90	68	1.5	70	1746
						Total kWh =		843,500

Table 7. Energy Efficient, Close Approach Evaporative Condenser Bin Hour Analysis of kW and kWh (HCFC-22 Reciprocating Compressor, Evaporative Condenser Sized for a 0.3 Total Application Factor)

Bin Temperature	Hours Year	Wet-Bulb	Load, Tons	Cond. Temp.	Comp. kW	Fan kW	Total kW	Total kWh
105/109	135	68	100	80	75	8.5	84	11323
100/104	315	66	100	79	74	8.5	82	25926
95/99	406	65	100	77	71	8.5	79	32141
90/94	508	63	100	75	68	8.5	76	38622
85/89	593	60	100	73	64	8.5	73	43223
80/84	636	58	100	71	61	8.5	70	44361
75/79	638	55	100	69	58	8.5	67	42498
70/74	645	52	100	66	53	8.5	62	39928
65/69	678	49	100	64	50	8.5	59	39843
d60/64	724	47	100	60	48	8.5	56	40906
55/59	781	44	75	66	40	2.8	43	33480
50/54	765	41	75	64	38	2.8	41	30993
45/49	669	38	75	62	35	2.8	38	25529
40/44	514	35	75	60	33	2.8	36	18404
35/39	361	31	75	60	33	2.8	36	12926
30/34	234	27	75	60	33	2.8	36	8379
25/29	94	23	75	60	33	2.8	36	3366
20/24	25	18	75	60	33	2.8	36	895
						Total kWh =		495,300

Table 8. HCFC-22 Net Refrigeration Effect Correction Factor (CF1) for Condensing and Wet Bulb Temperature Entering Wet Bulb Temperature °F (°C)

Condensing Temp °F (°C)	Condensing Press. psig kPa	50 (10.0)	55 (12.8)	60 (15.6)	62 (16.7)	64 (17.8)	66 (18.9)	68 (20.0)	70 (21.1)	72 (22.2)	74 (23.3)	76 (24.4)	78 (25.6)	80 (26.7)
70 (21.1)		0.48	0.37	0.26	0.21	0.16								
75 (23.9)		0.62	0.51	0.40	0.65	0.30	0.026	0.20	0.015					
80 (26.7)		0.76	0.66	0.55	0.51	0.46	0.41	0.36	0.30	0.25	0.19			
85 (29.4)	155.7 (1072.8)	0.96	0.86	0.74	0.70	0.65	0.60	0.55	0.49	0.44	0.38	0.32		
90 (32.2)	168.4 (1160.3)	1.11	1.01	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.54	0.49	0.43	0.37
95 (35.0)	181.8 (1252.6)	1.28	1.18	1.07	1.03	0.98	0.94	0.89	0.84	0.79	0.74	0.69	0.63	0.57
100 (35.7)	195.9 (1349.8)	1.43	1.33	1.24	1.20	1.16	1.12	1.07	1.03	0.98	0.93	0.87	0.82	0.76

Note: The table above will have a shaded section where the approach temperature is more than 15°F; with the following note "The shaded section should only be used for calculating TES nighttime energy savings. Maximum approach temperature should be less than 15°F."

This table may also be used for HFC-134a. Consult manufacturer.

Table 9. R-717 (Ammonia) Net Refrigeration Effect Correction Factor (CF1) for Condensing and Wet Bulb Temperature

		Entering Wet Bulb Temperature °F (°C)													
Condensing Temp °F (°C)	Condensing Press. psig kPa	50 (10.0)	55 (12.8)	60 (15.6)	64 (17.8)	66 (18.9)	68 (20.0)	70 (21.1)	72 (22.2)	74 (23.3)	76 (24.4)	78 (25.6)	80 (26.7)	82 (27.8)	84 (28.9)
65 (18.3)		0.38	0.27	0.14											
70 (21.1)		0.53	0.41	0.29	0.18										
75 (23.9)		0.68	0.56	0.44	0.34	0.28	0.22	0.16							
80 (26.7)		0.84	0.72	0.61	0.5	0.45	0.39	0.33	0.27	0.21					
85 (29.4)	151.7 (1045.2)	1.41	1.28	1.12	0.96	0.88	0.80	0.73	0.64	0.56	0.48				
90 (32.2)	165.9 (1143.1)	1.64	1.49	1.34	1.20	1.12	1.04	0.96	0.88	0.80	0.72	0.64	0.55	0.47	
95 (35.0)	181.1 (1247.8)	1.88	1.73	1.59	1.44	1.38	1.31	1.24	1.16	1.08	1.01	0.97	0.93	0.72	0.63
96.3 (35.7)	185.1 (1275.3)	1.93	1.81	1.64	1.51	1.44	1.38	1.31	1.25	1.17	1.08	1.0	0.90	0.79	0.69

Table 10. Net Refrigeration Effect Correction Factor (CF2) for System Suction Temperature for HCFC-22, HFC-134a, and Ammonia Refrigerants

	System Suction Temperature °F (°C)									
Suction Temperature °F	-40	-30	-20	-10	0	10	20	30	40	50
°C	(-40)	(-34)	(-28.9)	(-23.3)	(-17.8)	(-12.2)	(-6.7)	(-1.1)	(4.4)	(10.0)
HCFC-22/HFC-134a Capacity Factor (CF2)	0.76	0.79	0.82	0.85	0.88	0.92	0.95	0.97	1.0	1.03
R-717 (Ammonia) Capacity Factor (CF2)	0.82	0.85	0.88	0.91	0.94	0.97	1.0	1.03	1.06	1.09

Floating Head Pressure Calculation

To estimate energy savings, begin by determining the existing condensing temperature setpoint, and the minimum temperature the system can operate and still ensure positive refrigerant feed to the evaporators (with piping, valving, or other modifications).

Use the bin temperature method. Estimate existing and proposed minimum condensing temperatures. For each temperature bin, use Table 8 or 9 (depending on the refrigerant) to determine the change in condenser capacity; and the subsequent new condensing temperature, until the proposed minimum condensing temperature is obtained. From Table 4 or 5 (depending on the refrigerant) determine the change in compressor kW. Subtract the incremental energy usage of the condenser fans or other additional equipment that would not otherwise operate below the existing condenser setpoint.

Purging Calculations

Estimate energy savings by determining the change in condensing temperature after removing the non-condensable gas; then referring to Table 4 or 5.

Scaling Calculations

Determine energy savings by estimating the scale thickness; then using Graph 7 to determine the reduction in condenser capacity. Use Table 8 or 9 to determine the existing condenser application factor and the new application factor for the clean condenser. Determine the new condensing temperature using Table 8 or 9. Table 4 or 5 will provide the demand reduction based on the change in condensing temperature.

Proper Wet-Bulb Calculation

Calculate energy savings by estimating the existing wet-bulb at the air inlet and the new wet-bulb after installation of exhaust hoods or decking. Then use Table 8 or 9 (depending on the refrigerant) to determine the change in condenser capacity and the subsequent new condensing temperature. From Table 4 or 5 (depending on the refrigerant) determine the change in compressor kW.

Reducing Fan Horsepower Payback Calculation

For example, assume the incremental cost of the additional surface minus the reduction for the motor horsepower is approximately $18,000, and that the units will operate for 6,000 full-load hours per year. Also assume that the utility rate is $0.10 per kWh; and that the utility provides an incentive of $0.06 per kWh saved first year. The utility incentive would be approximately $5,100 ((55 - 36) hp x 0.746 kw/hp x 6,000 hours x $0.06/kWh). The annual energy saving is $11,400 ((55-36) hp x 0.746 kw/hp x 6,000 hours x $0. 10/kWh). The payback is $18,000 - $5,100/$11,400 = 13 months.

Example

A 100 ton HCFC-22 refrigeration system operating at +20°F suction temperature is to be installed in Las Vegas, NV. The peak wet-bulb temperature is 68°F. The contractor is considering the classic but inefficient design of 105°F condensing temperature; and an efficient close approach system with 80°F condensing temperature. The classical system has 4.7 kW of fan and pump load; and a 1.2 application factor. The close approach system has 8.5 kW of fan and pump load; and a 0.3 application factor. Both systems have two-speed motors. The contractor is installing an energy management system that will float the head pressure by cycling the condenser fans when the ambient dry-bulb temperature drops to 60°F. Tables 8 and 9 highlight the calculations.

Discussion

There are several items to notice from Tables 8 and 9. First, the close approach, energy efficient evaporative condenser has a 36 kW lower peak demand, and uses 348,200 less kWh, almost 40% less energy. At current utility rates in Las Vegas that will be an annual saving of over $25,000 per year. The system must be properly piped and installed to allow for floating head pressure.

The second item to notice is the change in system kW when the condenser fan cycles to half speed. In both cases the total system kW increased, for both the 75 and 100 ton loads. Also notice that the impact on system kW is far greater on the standard system. Also notice that cycling the fans at the same dry-bulb temperature results in a dramatically different kW demand impact.

Next, notice that with a 100 ton load floating the head pressure from 85°F to 60°F reduces compressor energy demand from about 83 kW to about 48 kW. When the incremental 5.7 kW energy demand of the fans is included, the net benefit is about 29 kW.

Table 11. Fan Capacity Correction Factors (FF)

Percent Fan Load	Percent HP Draw
100%	100%
90	97
80	94
70	91
60	88
35[(a)]	58

(a) For 2 speed motors

This analysis assumes the condenser capacity will decrease to roughly 58% when the fan speed is reduced to 35%, using a two speed energy efficient motor or a pony motor.

Endnotes

1. Nugent, D. 1993. "High-Efficiency Electric Technology Fact Sheet." Electric Power Research Institute, BR-102342.

2. "Survey of Manufacturers Representatives." 1992. Pacific Gas and Electric.

3. Carlyle Compressor Curves for a Discuss Compressor using HCFC-22 refrigerant.

4. Mycom and FES rotary screw compressor curves for ammonia refrigerant.

Reference

Electric Power Research Institute. 1993. *High-Efficiency Electric Technology Fact Sheet.* BR-102342.