COLMAC COIL
Manufacturing Inc.

Technical Bulletin

By Bruce I. Nelson, P.E., President, Colmac Coil Manufacturing, Inc.

COMPARING AIR COOLER RATINGS – PART 1:
Not All Rating Methods are Created Equal

Summary

Refrigeration air coolers (evaporators) are widely used to cool and circulate air in cold storage warehouses and food processing facilities. Manufacturers of air coolers publish cooling capacities based on differing assumptions and rating methods. It is important for refrigeration design professionals to understand these different rating methods and to apply them appropriately. In extreme cases, air coolers can be grossly undersized even though nominal catalog ratings appear to satisfy the calculated refrigeration load. The article illustrates the differences in these rating methods and highlights the importance of selecting air coolers using ratings suited to the operating conditions.

Background

Refrigeration air coolers (evaporators) are widely used to cool and circulate air in cold storage warehouses and food processing facilities. Manufacturers of air coolers publish cooling capacities based on differing assumptions and rating methods (Nelson 2010). In Europe, a number of manufacturers of commercial air coolers (i.e. coolers designed for use with R404a/R507) subscribe to the Eurovent certification program based on the European test standard EN 328 (EN 2002), however, no manufacturer is currently certified for industrial air coolers (i.e. coolers designed for use with ammonia refrigerant). In the U.S. the performance standard AHRI-420 exists (AHRI 2008), but no manufacturers participate in a certification program based on this standard. It is, therefore, important for refrigeration design professionals to understand the different rating methods being used and to apply them appropriately. In extreme cases, air coolers can be grossly undersized even though nominal catalog ratings appear to satisfy the calculated refrigeration load. The smaller size and lower first cost of air coolers which are inadvertently undersized due to misunderstood or misapplied ratings are seductively attractive to contractors and end users, however, the price difference will ultimately be more than paid for by the unsuspecting end user whose undersized air coolers cause lower-than-expected operating suction temperatures with associated increased energy consumption and loss of refrigerating capacity.

Air Temperature Change

As air passes across the fins of an evaporator coil, the temperature is reduced according to the following relationship (ASHRAE 2009):

$$\dot{q} = \dot{m} \cdot C_p \cdot (T_{ent} - T_{lvg})$$

where

$\dot{q}$ = cooling capacity (sensible only), Btu/h (kW)
$\dot{m}$ = mass flow rate of air, lbm/h (kg/s)
C_p = specific heat capacity of moist air, Btu/lbm F (kJ/kg C)
T_{ent} = dry bulb air temperature entering the coil (“air on” temperature), F (C)
T_{lvg} = dry bulb air temperature leaving the coil, F (C)

In a room being refrigerated by air cooling evaporators, the change in the temperature of the air (reduction) as it passes through the evaporator coils will equal the change in the temperature of the air (increase) as it circulates throughout the room. This means that in a well-designed cold room, the air temperature gradient found in the room will be roughly equal to and will be determined in large part by the air temperature change in the evaporator coils. By Equation (1) the magnitude of the air temperature change (gradient) in the room will be determined by the air mass flow rate through the evaporators. For example, if a relatively small air temperature gradient is desirable in a refrigerated room, then air coolers with relatively high air flow rate (i.e. high CFM/TR) for a given capacity must be selected.

Heat Exchanger Effectiveness

One well known method used to calculate the sensible cooling capacity of evaporators is the effectiveness method (Kays and London 1964). Heat exchanger effectiveness is defined as the ratio of the actual amount of heat transferred to the maximum possible amount of heat that could be transferred with an infinite area. This method is extremely useful because cooling capacity can be calculated directly knowing only the dimensional characteristics of the coil and the initial temperature difference (entering air temperature minus the evaporating temperature). This initial temperature difference is referred to as "DT1" (or "TD") in the refrigeration industry. Sensible cooling capacity is calculated as follows:

$$\dot{q} = \dot{m} \cdot C_p \cdot \epsilon \cdot \left(T_{ent} - T_{evap}\right) = \dot{m} \cdot C_p \cdot \epsilon \cdot DT1$$

where

$\dot{q}$ = cooling capacity (sensible only), Btu/h (kW)
$\dot{m}$ = mass flow rate of air, lbm/h (kg/s)
C_p = specific heat capacity of moist air, Btu/lbm F (kJ/kg C)
ϵ = effectiveness = $\left(T_{ent} - T_{lvg}\right)/\left(T_{ent} - T_{evap}\right)$
T_{ent} = dry bulb air temperature entering the coil ("air on" temperature), F (C)
T_{lvg} = dry bulb air temperature leaving the coil, F (C)
T_{evap} = average refrigerant evaporating temperature, F (C)

For a given sized coil operating with constant air flow rate, the effectiveness can be considered constant over the small operating temperature ranges typical of refrigeration applications, and therefore, capacity can be considered to be proportional to the ratio of DT1. Hence, if evaporator coil sensible capacity is known for a given DT1, then capacity at a new initial temperature difference, DT1', can be found simply by multiplying the original capacity by the ratio DT1'/DT1. For example, a refrigeration air cooler has a rating of 10 TR at a DT1 of 10F. The capacity of the same cooler operating with a new DT1 of 12F will be very close to 10 x 12/10 = 10 x 1.2 = 12 TR.

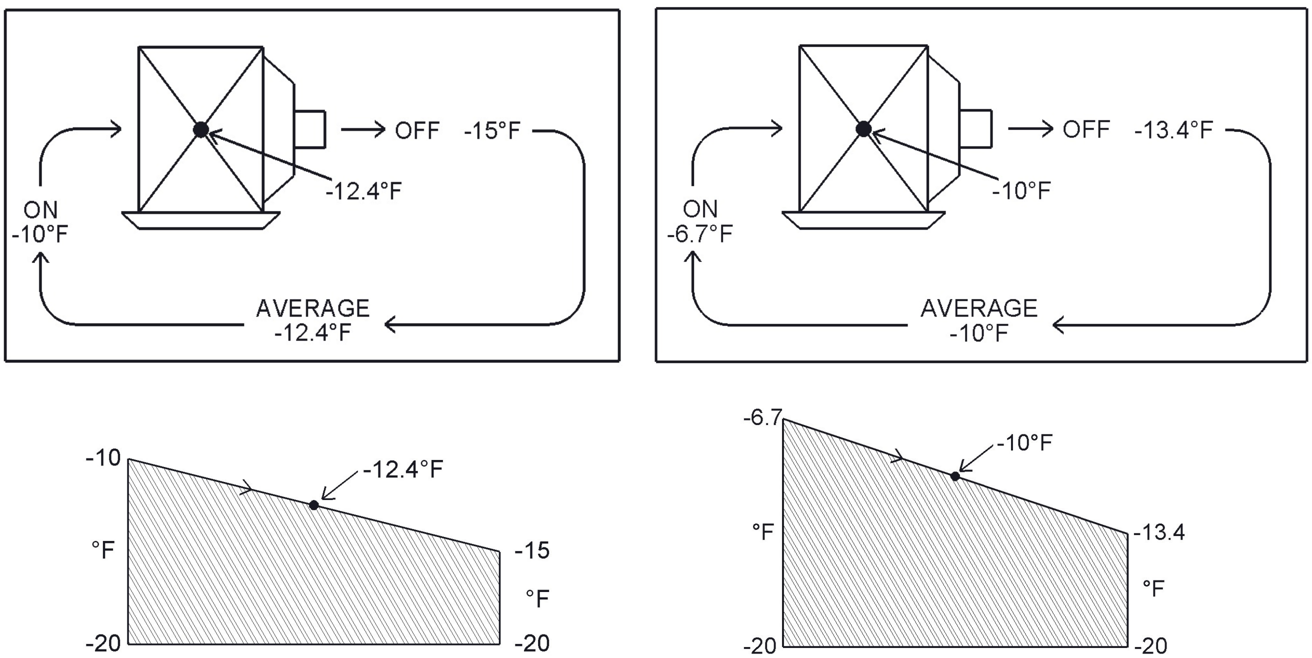
Average Room Temperature and DTM Ratings

Control of the refrigeration system is normally accomplished by maintaining room air temperature, that is, compressors and coolers are cycled on or off depending on whether room temperature is rising or falling. Location of the air temperature sensing device relative to the location of evaporators will affect evaporator performance since a temperature gradient always exists in the room and, as seen above, evaporator performance is determined by the air on temperature (i.e. by DT1). Evaporators located high in the room (mounted on the ceiling, for example), will be exposed to the highest air temperature in the room and operate with the largest DT1. Conversely, floor mounted evaporators will be exposed to the coldest air in the room and operate with the smallest DT1.

For the specific case where; 1) air coolers are ceiling mounted (i.e. operate at the warmest location in the room), and 2) the system control temperature sensor is mounted at a location where it will sense the *average* room temperature (i.e. at the midpoint elevation in the room), manufacturers of air coolers publish ratings based on *mean* (average room) temperature difference. This average temperature difference is termed "DTM". DTM ratings for the same air cooler will always be higher than DT1 ratings since the effective initial temperature difference seen by the evaporator coil is higher by approximately ½ of the air temperature change. Figure 1 below illustrates how air temperature changes as it passes through an evaporator at two different operating conditions. Note that airflow is held constant for both operating conditions. The first condition with DT1 = 10F is shown as Figure 1(a). The second condition with DTM = 10F is shown as Figure 1(b). Because the DTM = 10F condition has the larger *initial* temperature difference of -6.7 – (-20) = 13.3F, the cooling capacity and air temperature change are significantly larger than for the DT1 = 10F condition. It is interesting how the same evaporator can produce more cooling capacity simply by redefining "temperature difference"!

FIGURE 1
Temperature Profiles for DT1 vs DTM

(a) DT1 = 10F (Air On) Temp Difference **(b) DTM = 10F (Average) Temp Difference**



DT1 LMTD = 7.2 F DTM LMTD = 9.6 F

Although DTM ratings may useful in the specific case of ceiling mounted air coolers (equipment size and first cost can be reduced), refrigeration system designers must be careful to recognize the limitations of this rating system and avoid the mistake of misapplying DTM ratings. Whenever air coolers are installed at a location in the room where the air on temperature to the coil is less than the highest temperature in the room, DTM ratings should not be used. This would be the case with cooler inlet air ducted from some lower temperature point in the room, or with floor mounted coolers.

DT1 ratings used with actual anticipated air on temperature will always result in accurate ratings and correct air cooler selections. This method for air cooler selection is conservative and recommended whenever air on temperature to the coil is less than the maximum found in the room or process.

Converting DT1 to DTM Air Cooler Ratings

Equations (1) and (2) above were used to derive the relationships shown in Equations (3) and (4) below which can be used to convert from known a DT1 air cooler rating to a new DTM rating for the same cooler.

$$\dot{q}_{DTM} = \frac{\dot{q}_{DT1} \cdot \frac{DTM}{DT1}}{\left(1 - \frac{\dot{q}_{DT1}}{2 \cdot 60 \cdot C_p \cdot \rho \cdot \dot{V} \cdot DT1}\right)} \quad \text{(IP)} \qquad (3)$$

$$\dot{q}_{DTM} = \frac{\dot{q}_{DT1} \cdot \frac{DTM}{DT1}}{\left(1 - \frac{\dot{q}_{DT1} \cdot 3600}{2 \cdot C_p \cdot \rho \cdot \dot{V} \cdot DT1}\right)} \quad \text{(SI)} \qquad (4)$$

where

$\dot{q}_{DTM}$ = capacity at mean (room) temperature difference, Btu/h (kW)
$\dot{q}_{DT1}$ = capacity at initial (air on) temperature difference, Btu/h (kW)
DT1 = initial temperature difference = Air On Temp - Evap Temp, F (C)
DTM = mean (room) temperature difference = Ave Room Temp – Evap Temp, F (C)
C_p = air specific heat, Btu/lbm F (kJ/kg C)
ρ = air density, lbm/ft3 (kg/m3)
$\dot{V}$ = actual volumetric air flow rate, ft3/min (m3/h)

Example:

An air cooler has a DT1 rating of 120,000 Btu/h at DT1 = 10F and -10F air on temperature. The cooler has a published airflow rating of 18,850 CFM. Assume the coil is operating with average air density = 0.0883 lbm/ft3, and average air specific heat = 0.24 Btu/lbm F. Note this is the same cooler shown in Figure 1 above.

Find the DTM rating for the same cooler with DTM = 10F.

From Equation (3):

$$\dot{q}_{DTM} = \frac{120{,}000 \cdot \frac{10}{10}}{\left(1 - \frac{120{,}000}{2 \cdot 60 \cdot 0.24 \cdot 0.0883 \cdot 18{,}850 \cdot 10}\right)} = 160{,}050\ Btu/h$$

As is seen from the example, DTM ratings are typically significantly higher than DT1 ratings for the same air cooler operating under the same conditions. In the case of the example, the DTM rating is +33% greater than the DT1 rating!

Note that the above equations apply only to sensible capacity calculations and ratings and do not account of the effects of latent cooling on coil performance and temperature change. The effects of latent load on coil performance and ratings are covered in the following sections.

Latent Load And Sensible Heat Ratio (SHR)

Whenever cooling coil surfaces operate at temperatures below the dewpoint of the air being cooled, water vapor in the airstream is condensed to liquid (at temperatures above 32F (0C)) or deposited to form frost (below 32F (0C)). The cooling effect associated with this dehumidification of the airstream is termed “latent” cooling. The sum of the sensible cooling load and latent cooling load is termed the “total” load. The ratio of the sensible cooling load divided by the total cooling load is called the Sensible Heat Ratio (SHR) and defines the slope of the air process line on a psychrometric chart.

$$SHR = \frac{Sensible\ Cooling\ Load}{Sensible\ Cooling\ Load + Latent\ Cooling\ Load} \qquad (5)$$

Accurate prediction of the refrigeration load, both sensible and latent components, is critical to proper refrigeration system equipment selection and successful operation. Various types of sensible cooling loads must be anticipated and included in the calculation, such as: lighting, electric motors, forklifts, product cooling/freezing, transmission of heat through walls, ceilings, and floors, and cooling of infiltration air. Latent cooling loads are present whenever moisture is added to the air in the refrigerated space. Sources of introduced moisture typically include: infiltration air, respiring food products, surface moisture on products, packaging, and other objects entering the space, human respiration, and humidification equipment (above freezing).

The SHR determined from the load calculation will come to equilibrium with the SHR of the air passing through the air cooler evaporator coil. In general, as air temperature decreases the amount of water vapor held in air decreases by the law of partial pressures, and the minimum possible SHR increases.

Relative humidity of the refrigerated space can be predicted by plotting the air process line on a psychrometric chart with the end point plotted on the saturation curve at the predicted coil surface temperature, and the air process line extending from left to right at a slope equal to the SHR. The intersection of this line with a vertical line drawn through the entering air dry bulb temperature indicates the relative humidity of the air entering the coil. Table 1 below shows typical Sensible Heat Ratios for various air temperatures at 95% air on relative humidity.

TABLE 1
SHR FOR 95% RH AIR ON AND DT1 = 10F AT VARIOUS TEMPERATURES

Room Temperature, F (C)	SHR
45 (7.2)	0.55
32 (0)	0.66
10 (-12.2)	0.83
0 (-17.8)	0.89
-10 (-23.3)	0.93
-30 (-34.4)	0.97

Impact of Latent Load (SHR) on Air Cooler Ratings

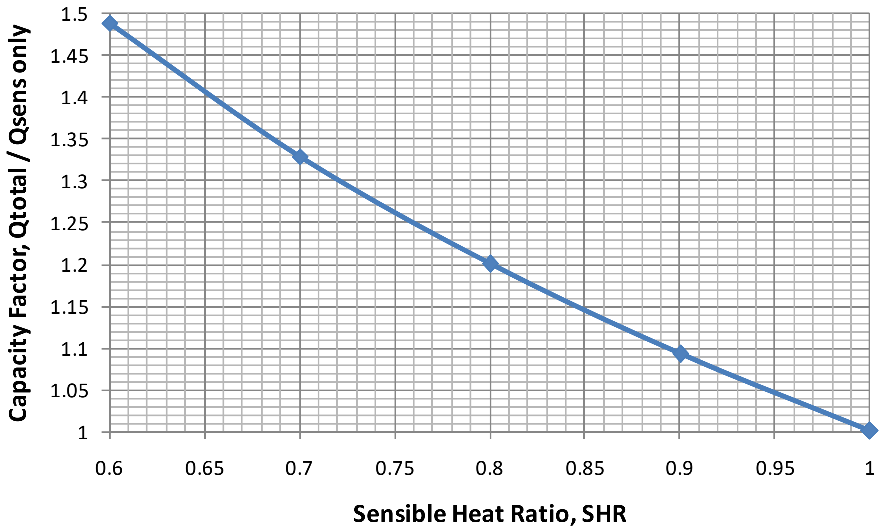
Evaporator coils are typically constructed of plate fins bonded to tubes. Fins are referred to as “secondary” surface while tubes are referred to as “primary” surface. For purposes of calculating evaporator performance, primary surface is considered to be 100% effective in its contribution to the total heat transfer surface area while secondary surface has a surface effectiveness less than 100% due to the change in surface temperature from the root to the tip of the fin.

The capacity of the evaporator surfaces to transfer mass (condense water, or deposit frost, from the airstream) is a function of the difference in water vapor pressure between the coil surface and air stream and the surface mass transfer coefficient. The mass transfer process is much more “thermally effective” than the sensible heat transfer process, that is, the heat flux through the evaporator surfaces during the mass transfer process is extremely high (AHRI 2001). Consequently, if the surface effectiveness of the coil were to remain constant, the increase in the evaporator cooling capacity during combined sensible and latent cooling would be equal to the sensible cooling capacity divided by the SHR, as follows:

$$Total\ Cooling\ Capacity_{Ideal} = \frac{Sensible\ Cooling\ Capacity}{SHR} \qquad (6)$$

However, the increase in heat flux through the fin surfaces has the effect of decreasing fin efficiency and overall surface effectiveness due to an increase in the fin surface temperature gradient (Xia & Jacobi 2005). The result is a slightly lower total cooling capacity than that predicted by Equation (6). Using a computer model developed to accurately calculate fin efficiency and surface effectiveness for both sensible and combined sensible and latent heat transfer, a prediction of the increase in evaporator coil performance as a function of SHR was made. Results of the predicted capacity increase as a function of SHR for an ammonia refrigeration evaporator coil operating over a wide range of room temperatures (+35F to -30F) and having typical fin spacing and geometry with DT1 = 10F are shown in Figure 2 below.

FIGURE 2
Total Cooling Capacity Factor vs SHR



Air cooler ratings which include latent cooling will appear higher (in some cases significantly higher) than all sensible ratings. Care must be taken, therefore, to correctly predict the cooling load SHR and the resulting relative humidity in the refrigerated space. From the above it should be apparent that selecting an air cooler with a rating based on a relatively high room relative humidity (SHR less than 1.0) for a room with an actual SHR equal to or close to 1.0 will result in undersized air coolers.

For example, a long term cold storage warehouse is designed for +0F (-17.8C) room temperature with a calculated SHR nearly equal to 1.0 (i.e. packaged products and minimal infiltration). From Table 1, the SHR for +0F air temperature and 95% relative humidity would be 0.89. From Figure 2, an air cooler rated on a total cooling basis at +0F and 95% air on relative humidity would show a nominal capacity +11% greater (capacity factor = 1.11) than a sensible only rating. In this case, therefore, air coolers selected using ratings based 95%rh air on would be significantly undersized.

Conclusions

U.S. air cooler manufacturers have traditionally published capacity ratings based on SHR = 1.0 (all sensible) and DT1. European manufacturers typically include latent cooling in their air cooler ratings, indicated by an air on relative humidity typically between 85% and 95%. European manufacturers also publish ratings based on either DT1 or DTM, or both. The discussion above illustrates the differences in these rating methods and highlights the importance of selecting air coolers using ratings suited to the operating conditions. Misapplication of DTM and/or total cooling ratings can result in severely undersized air coolers and the consequent failure of the refrigeration system to perform to energy efficiency and cooling capacity expectations.

Bibliography

AHRI 2001. Standard 410. “Forced-Circulation Air-Cooling and Air-Heating Coils”. Air-Conditioning Heating & Refrigeration Institute. Arlington, VA. Section 6.2.1.

AHRI 2008. Standard 420. “Performance Rating of Forced-Circulation Free-Delivery Unit Coolers for Refrigeration ”. Air-Conditioning Heating & Refrigeration Institute. Arlington, VA

ASHRAE 2009. 2009 ASHRAE Handbook - Fundamentals. *American Society of Heating Refrigerating and Air-Conditioning Engineers.* Atlanta, GA. Chap. 4, pp 4.18.

EN 2002. Standard EN 328. “EN 328 - Heat exchangers - Test procedures for establishing the performance of forced convection unit air coolers for refrigeration”. European Norm

Kays, W.M., A.L. London. 1964. Compact Heat Exchangers - Second Edition. *McGraw-Hill Book Company*. Chap. 2, pp. 15-24.

Xia, Y., A.M. Jacobi. 2005. “Air-side Data Interpretation and Performance Analysis for Heat Exchangers with Simulataneous Heat and Mass Transfer: Wet and Frosted Surfaces*”. International Journal of Heat and Mass Transfer.* Vol. 48 (2005) 5089-5102.

For more information contact Colmac Coil Manufacturing, Inc.
www.colmaccoil.com | (800) 845-6778 | (509) 684-2595
P.O. Box 571, Colville, WA. 99114-0571