

UNITARY A/C: THE INFLUENCE OF PRODUCT DESIGN AND AIR FLOW RATES ON SENSIBLE HEAT RATIO

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ABSTRACT

Concerns continue to persist that higher efficiency unitary A/C designs are developed at the expense of higher Sensible Heat Ratios (SHRs) and degraded dehumidification. In this paper, SHR is investigated from a fundamental psychrometric standpoint. The factors influencing SHR are identified and related to the system design variables that influence seasonal energy efficiency ratio (SEER). In the end, it is shown that SEER has very little impact on SHR, while indoor airflow is the major variable, which was probably well understood by the earlier industry pioneers who wisely set the limiting rating condition at 450 scfm per ton (37.5 scfm per 1000 Btuh) of rated capacity. (Note: “scfm” stands for “standard cfm,” the air flow rate for the standard air density of 0.075 lbm/ft³).

Key Words: *air conditioners, evaporators, latent capacity*

1 INTRODUCTION

On-going concerns that higher SEER designs have led to higher SHR levels and sacrifices in dehumidification have been generally accepted by the HVAC community and others. ARI has recently published results of a statistical study (Amrane and Hourahan 2002) that attempt to dispel some of the fears. Figure 1 from the ARI paper shows that, considering all small unitary products in 2001, there is no general relation between SHR-A and either SEER or EER-A. ARI Standard 210/240 (ARI 1994) places an upper limit on the indoor airflow that can be used with ARI-rated products. The present paper will show that this limit has had a major impact on product SHRs and is a principal explanation for the results shown in Figure 1.

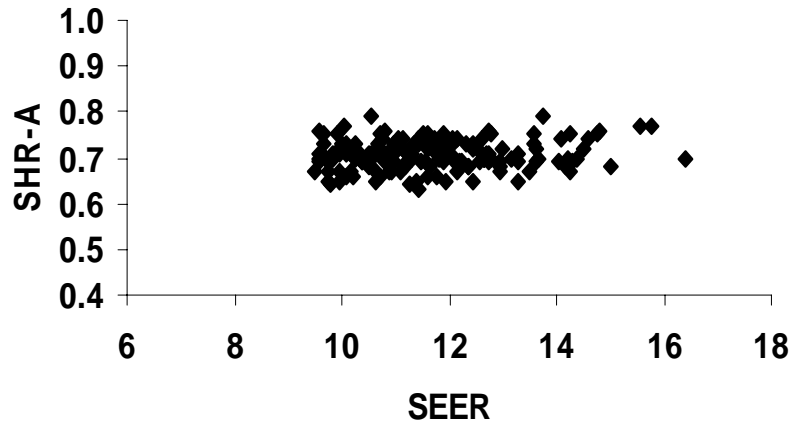


Fig. 1. ARI Test Results for Sensible Heat Ratio at 95/80/67 vs. SEER (Amrane and Hourahan 2002).

2 PSYCHROMETRIC ANALYSIS

The basic relations between scfm/ton and SHR can be described by only a few simple relations. The first equation relating tons cooling to air enthalpy change is

$$\text{Tons} = \text{scfm} * \rho_{\text{std}} * 60 * (\text{Hin} - \text{Hout}) / 12000 \quad (1)$$

where scfm = evaporator air flow rate, adjusted to standard air
density, cu. ft. /min

ρ_{std} = standard air density, 0.075 lb/cu. ft.

Hin = enthalpy of air entering evaporator, Btu/lb dry air

Hout = enthalpy of air leaving evaporator, Btu/lb dry air

Then

$$\text{Hout} = \text{Hin} - \frac{266667}{\text{scfm/ton}} \quad (2)$$

Equation 2 says that for a given entering air condition and a known airflow rate, the leaving air condition must lie along a line on the psychrometric chart defined as Hout. For the standard ARI entering air case of 80F dry-bulb, 67F wet-bulb, Hin = 31.519 Btu/lb dry air. Figure 2 shows a simple psychrometric chart with enthalpy lines for Hin and several values of Hout, corresponding to 300 to 450 scfm/ton). Hout can be anywhere along the line for the given scfm/ton. The maximum possible SHR would be for the leaving air condition to be at the saturation condition, which would require an infinite size evaporator coil as explained later.

Sensible Heat Ratio (SHR) is the ratio of the sensible cooling capacity to the total cooling capacity (sensible plus latent). This can be expressed by the following:

$$\text{Qsens} = \text{scfm} * \rho_{\text{std}} * 60 * (\text{Hmid} - \text{Hout}) / 12000 \quad (3)$$

$$\text{Qlatent} = \text{scfm} * \rho_{\text{std}} * 60 * (\text{Hin} - \text{Hmid}) / 12000 \quad (4)$$

$$\text{Qtotal} = \text{scfm} * \rho_{\text{std}} * 60 * (\text{Hin} - \text{Hout}) / 12000 \quad (5)$$

where Qsens = Sensible cooling capacity, Btu/hr

Hmid = enthalpy at *entering air dry-bulb and leaving air*

humidity ratio, Btu/lb dry air
 Q_{latent} = Latent cooling capacity, Btu/hr
 Q_{total} = Total cooling capacity, Btu/hr

Then

$$\text{SHR} = \frac{H_{\text{mid}} - H_{\text{out}}}{H_{\text{in}} - H_{\text{out}}} \quad (6)$$

Figure 2 illustrates this.

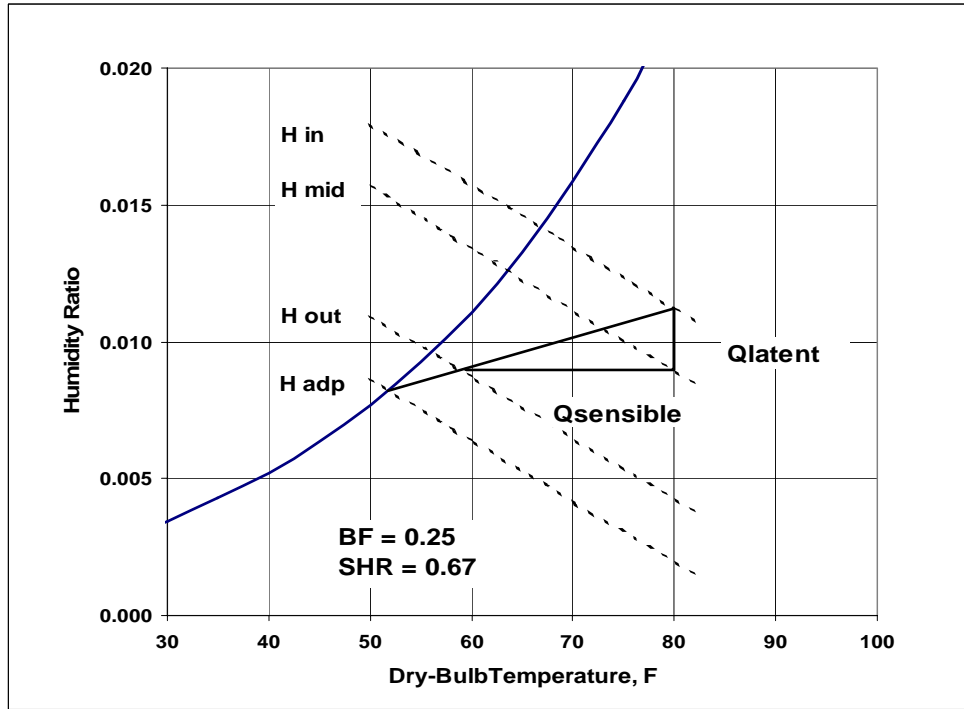


Fig. 2. Psychrometric Chart for Latent vs. Sensible Capacity

As H_{out} increases, it is clear that Q_{latent} will decrease and SHR will increase. Given the ARI entering air rating condition of 80F/67F and the maximum allowable rated airflow of 450 scfm/ton at the capacity rating condition (95F outdoor temperature), H_{out} calculates out to 25.59 Btu/lb dry air, and the resulting SHR for leaving air condition at saturation would be 0.88. Restated, the absolute maximum SHR of any air conditioner rated according to ARI 210/240 will be 0.88 irrespective of any of its design attributes, or its efficiency rating. If the air conditioner were rated at 350 scfm/ton, the resulting SHR for leaving air condition at saturation would decrease to 0.77.

In actual practice the evaporator leaving air condition cannot be fully saturated, due to the fact that cooling coils have limited surface areas. The inability of a heat exchanger to achieve completely saturated exit air can be described by a bypass factor BF. This concept has been in use for many decades and is described in detail in the literature (Kuehn et al 1998, Sherif 2002). With this concept, a fictitious fraction BF of the air stream is assumed to be totally unaffected by the

coil, i.e., “bypassed.” The remaining fraction, $1 - BF$, is brought to complete saturation at an apparatus dew point temperature T_{adp} . These two air streams then re-mix to form the actual exit air state. This is shown in Fig. 3.

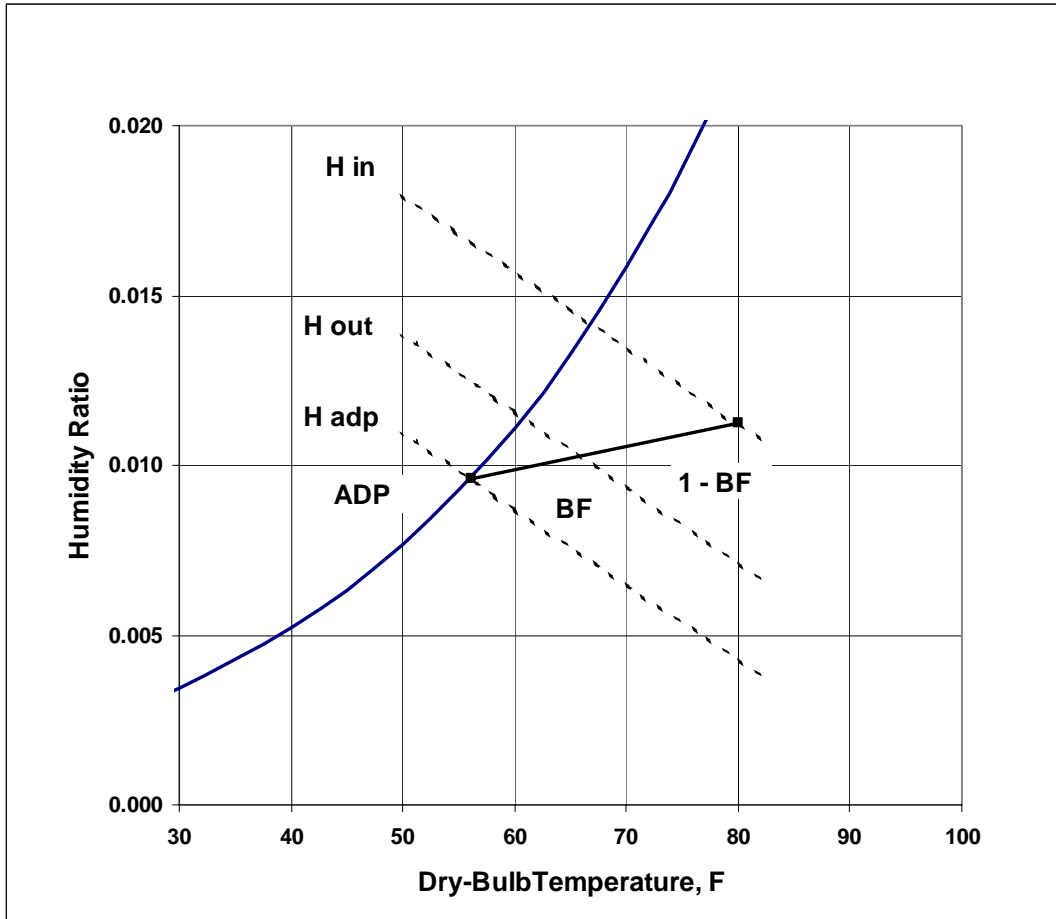


Fig. 3. Psychrometric Chart Defining Bypass Factor

By the definition of the bypass factor,

$$Q_{total} = scfm * \rho_{std} * 60 * (1 - BF) * (H_{in} - H_{adp}) / 12000 \quad (7)$$

From equations (5) and (7),

$$BF = \frac{H_{out} - H_{adp}}{H_{in} - H_{adp}} \quad (8)$$

Figure 4 shows SHR as a function of airflow rate and BF, using the above equations and additional air property relations.

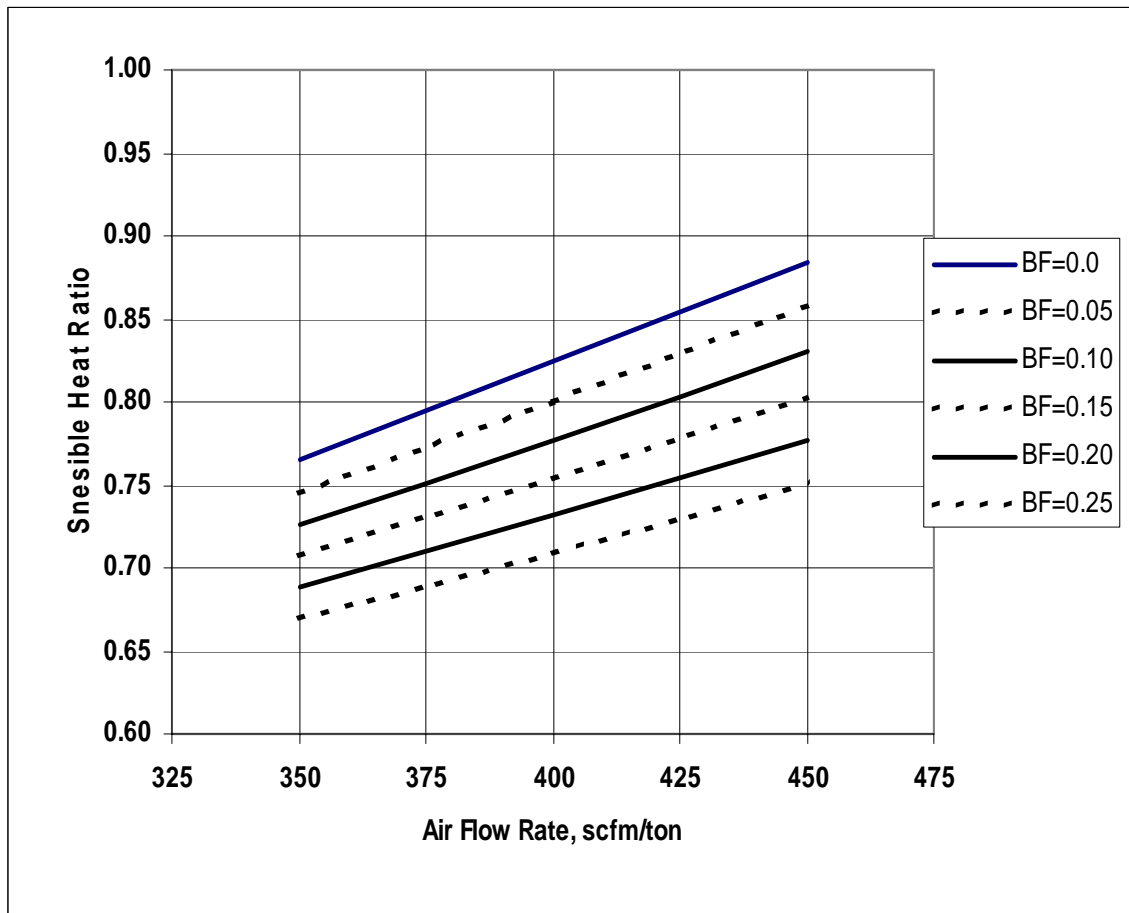


Fig. 4. Sensible Heat Ratio versus Evaporator Air Flow Rate and Bypass Factor

Note that

- SHR rises with increasing scfm/ton. This is to be expected since H_{out} increases, forcing H_{mid} closer to H_{in} ;
- SHR drops with increasing BF. This seems to be contradictory, i.e. one would expect low BFs to give better coil contact and more dehumidification. However, a larger BF leads to a lower ADP (implying a colder coil surface), which results in the coil cooling and dehumidifying a reduced quantity of air, such that the unchanged total mixed air flow quantity exits at a slightly higher dry bulb temperature with reduced moisture content, thus the lower SHR.

3 BYPASS FACTOR EVALUATION

The numerical value of the bypass factor BF is influenced by two main factors:

1. The total heat transfer surface area, principally the air-side area, as determined by
 - Coil face area
 - Fin density

- Number of rows
- 2. The air flow rate, principally the local velocity

To illustrate these effects, a proprietary air conditioner system design program was run with numerous variations of a basic 3-ton split-system AC unit design, at the ARI capacity rating condition of 95F outdoor and 80F/67F indoor entering air conditions. The basic unit design had an evaporator with a face area of 4 sq. ft., a fin density of 14 fpi and 3 rows. The compressor was a conventional scroll type and the condenser had 1 row, with a face area of 12.4 sq.ft. and an air flow rate of 2800 scfm. With an evaporator airflow rate of 1200 scfm (400 scfm/ton) and a face velocity of 400 ft/min., the unit EER is 11.5 Btu/Wh at ARI point A (95F/75F, 80F/67F). The design program was run such that, for each change in the evaporator coil geometry, the system was recharged and the compressor re-sized to achieve a system cooling capacity of exactly 3 tons. In addition, the evaporator fan motor power was recalculated for changes in airflow rate and coil pressure drop, assuming a constant fan efficiency and external static pressure (0.3 in. H₂O).

Figure 5 summarizes results of the design runs, showing BF as a function of scfm/ton and coil face area, for a *fixed number of rows* (3). The original coil design (4 sq. ft., 400 scfm/ton) has a BF of 0.14 and an SHR of 0.76.

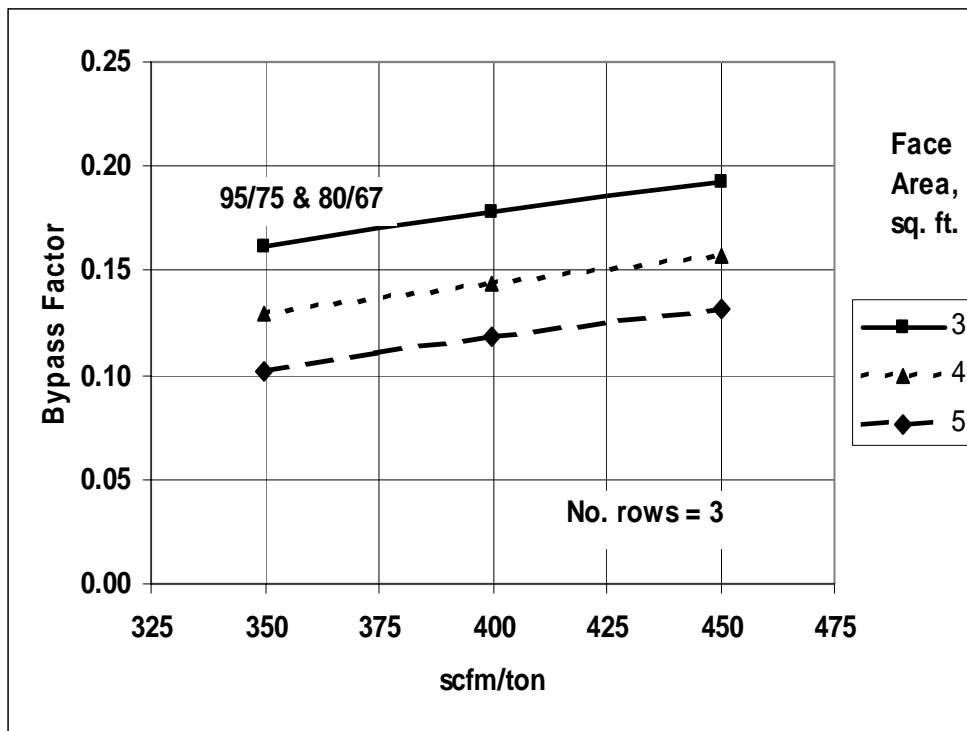


Fig. 5. Bypass Factor versus Evaporator Air Flow Rate – Fixed Number of Rows. Note that BF rises with increasing scfm/ton and BF drops with increasing face area.

The previous runs were made with a constant evaporator fin density (14 fpi). The following table shows results of varying the fin density, for a fixed face area, number of rows and scfm/ton:

fpi	BF
11	0.189
14	0.128
17	0.085
20	0.057

Overall, the typical bypass factor for practical unit designs ranges from about 0.10 to 0.25. As shown in Fig. 4, the calculated SHR for a limiting airflow of 450 scfm/ton and a low bypass ratio (for that airflow) of 0.15, is 0.80, while the calculated SHR for a lower practical airflow of 350 scfm/ton and a high bypass ratio of 0.25 is 0.67. These boundary values agree fairly well with the previously referenced ARI study (Amrane and Hourahan 2002) of actual industry test data for SHR (Fig. 1).

Figure 6 shows the corresponding SHR values for the previously presented evaporator designs.

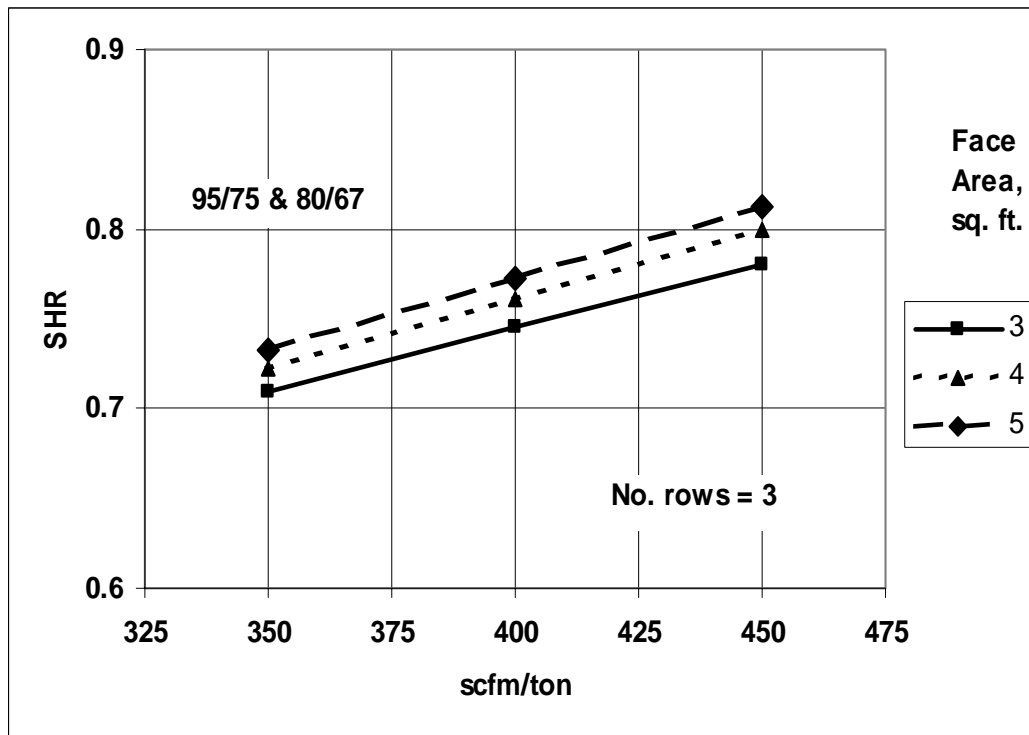


Fig. 6. Sensible Heat Ratio versus Evaporator Air Flow Rate – Fixed Number of Rows

Compared to the relative large changes shown in Fig. 5 for bypass factor, Fig. 6 shows relatively small changes in SHR. This can best be understood by looking at Figs. 7 and 8, which are compilations of Figs. 2 and 3, and demonstrates graphically that a relative large change in BF (0.07 in Fig. 7, 0.25 in Fig. 8) results in only a small change in SHR because of the inherent psychrometric properties of moist air.

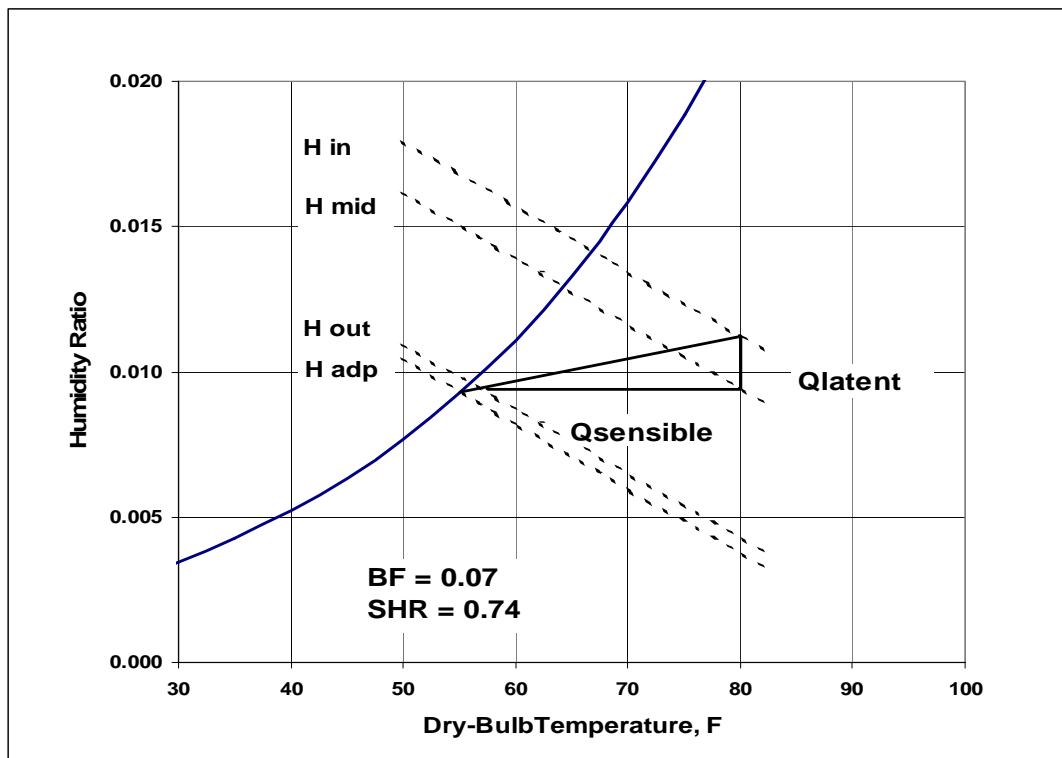


Fig. 7. Psychrometric Chart – Low Bypass Factor Coil

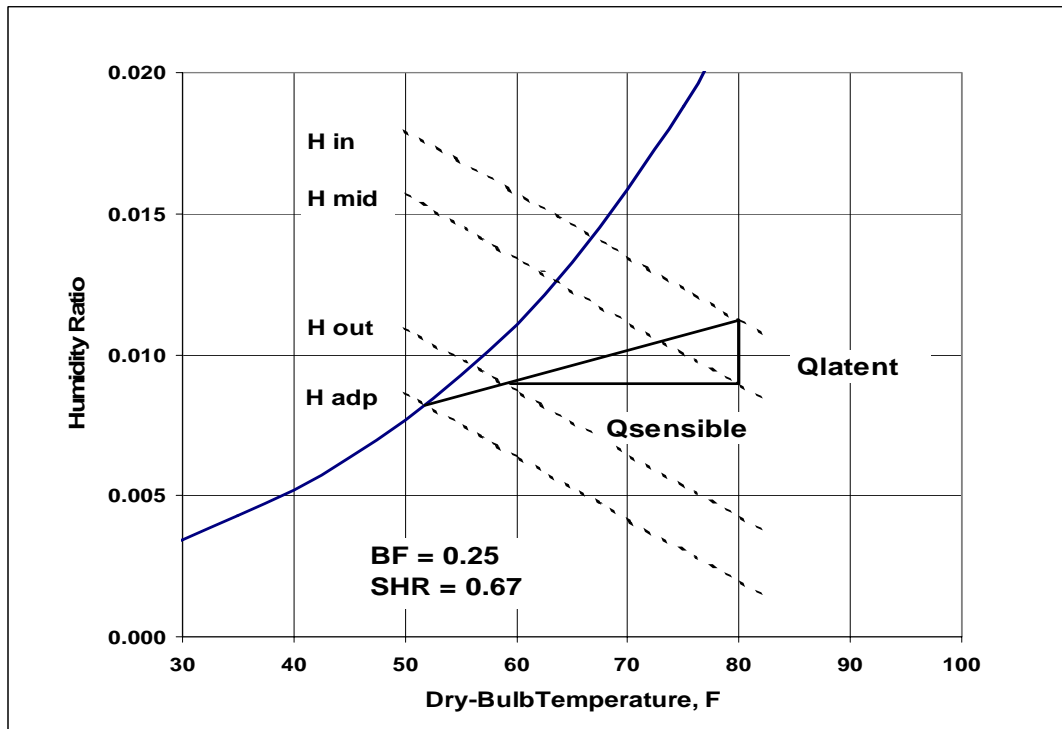


Fig. 8. Psychrometric Chart – High Bypass Factor Coil

From all of the preceding analysis, it should be clear that the major variable in arriving at an air conditioning systems sensible heat ratio is airflow, and that an installer should insure that the cooling airflow of the installed system does not exceed 450 scfm/ton.

4 MODIFIED EVAPORATOR ENTERING AIR STATE

Earlier in this discussion, the term “entering air condition” was used, and the “rating condition” of 80F/67F was given. In an actual application, this entering air condition to the coil becomes the house return air plus any change as it travels thru the return air ductwork. Again, the earlier industry pioneers who established this rating condition seem to have given it considerable thought. 80F/67F refers to 80F dry bulb temperature and 67F wet bulb temperature, or 50% relative humidity. 80F dry bulb lies at the upper end of the summer comfort zone as defined by ASHRAE. If it is assumed that a homeowner actually operates his/her system with an entering coil air condition of 77F/66F (55% rh), then a system rated and applied with 400 scfm/ton and a bypass ratio of 0.12 would see the SHR decrease from a rated 0.72 to an operating 0.66. If we were to further assume a high latent load in the house such that the relative humidity increased to 60%, then the entering air condition would be 77F/67F, and the SHR would decrease further to 0.62. From this analysis it is clear that any given air conditioner design will operate at a reduced SHR in response to an increase in indoor relative humidity.

Figure 9 shows the SHR values for the new indoor entering air operating condition (77F/67F) for the various evaporator design options. Note that the SHR values are indeed substantially lower than the corresponding ones shown in Fig. 6 for the 80F/67F entering condition. It should be noted that the scfm/ton values are based on the tonnage at the standard rating point (95F/75F, 80F/67F), not at the modified entering air state.

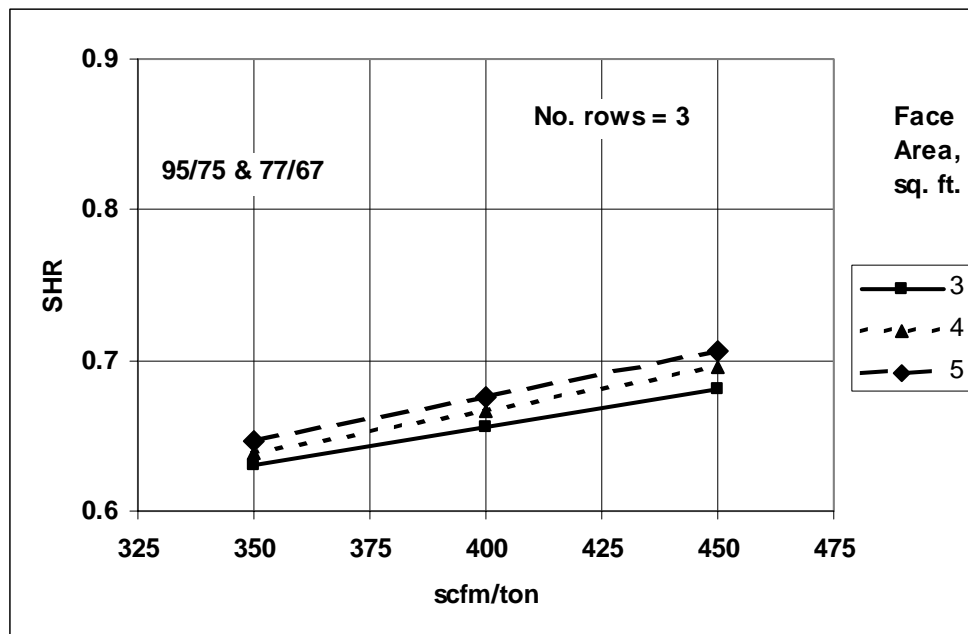


Fig. 9. Sensible Heat Ratio versus Evaporator Air Flow Rate – Fixed Number of Rows – Modified Entering Air State

5 MODIFIED OUTDOOR AIR STATE

All of the discussion to this point has been based on the ARI capacity rating condition of 95F outdoor temperature. As the outdoor temperature changes, the systems capacity will typically increase 0.53% for each degree F of outdoor temperature decrease. In other words, as the outdoor temperature changes from 95F to 82F, the capacity would increase by about 7%. This in turn would effectively decrease the indoor scfm/ton from 400 to 372, at a constant bypass ratio. Thus, our original SHR of 0.72 would change to 0.69 with the decrease in outdoor temperature. Recalculating our boundary ranges of 450 scfm/ton at 0.10 bypass and 350 scfm/ton at 0.15 bypass for the effect of decreasing the outdoor temperature from 95F to 82F, yields airflows of 418 and 325 scfm/ton respectively, and gives SHR values of 0.76 and 0.62, again agreeing almost exactly with the ARI study of actual industry test data for SHR. Figure 10 shows the SHR values for the decreased outdoor temperature condition versus the various evaporator design options.

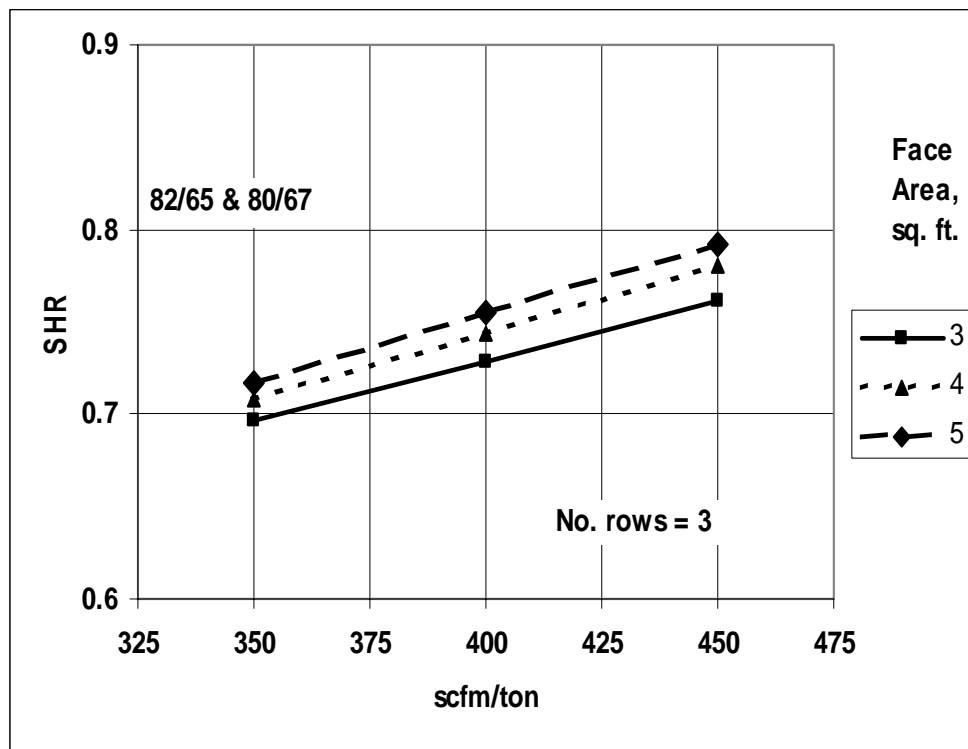


Fig. 10. Sensible Heat Ratio versus Evaporator Air Flow Rate – Fixed Number of Rows – Modified Outdoor Air State

Again, the SHR values are slightly lower than those for the standard rating point (95F/75F, 80F/67F) because the scfm/ton values are based on the lower capacity of the unit at the standard rating point, not the increased capacity associated with the reduced outdoor air temperature.

6 CONCLUSIONS

- Evaporator scfm/ton has a very significant effect on the evaporator bypass factor (BF), and sensible heat ratio (SHR);
- Evaporator design has a significant effect on BF, and some attendant effect on SHR;
- Although product designs can be devised with SHR values in excess of 0.8, practical limits on hardware size generally limit this;
- In order to keep unit SHRs below 0.8, it is *extremely important to insure that evaporator airflow rates do not exceed the ARI limit of 450 scfm/ton*. This applies as well to multiple-capacity units with air flow adjusted appropriately as the tonnage changes;
- There is little basis for the concern that higher SEER designs tend to have higher SHR values. Fundamental psychrometric limitations and the ARI restriction on evaporator air flow to less than 450 scfm/ton have insured that such products *when properly applied* do not have SHR above about 0.8, which is an acceptable upper limit for the ARI rating point.

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