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INTRODUCTION

The increasing emphasis on air conditioning product efficiency has created a COP (Btu/watt-hr) race and placed tremendous pressure on the industry. Overall awareness of the energy situation is creating a better informed consumer, who demands not only comfort from his air conditioning system, but high efficiency as well. He is now willing to make an additional first cost investment to secure long term operating cost advantages. The system design engineer has a strict performance target to add to his list of design restraints.

In designing an air conditioning system, knowledge of the performance of all possible components is required. Component modeling coupled with limited testing provides this information. The final problem is to design the lowest cost unit which meets the performance and capacity constraints while using available components.

A wide range of compressors is available. The design engineer must examine literally dozens of possible compressors before arriving at a final selection.

Four major factors control system performance. These are:

1. Evaporating temperature.
2. Condensing temperature.
3. Compressor efficiency.
4. System losses.

In addition, the requirements of various certifying and regulating agencies must be considered.

In the press for higher efficiency, the system designer must reevaluate the major factors. As these parameters are changed, the system designer finds the compressors previously used in a given system no longer proper. This will place pressure on the compressor designer to produce different displacements than are presently available.

The authors of this paper are presently involved in such a system reevaluation. This paper has been prepared in order to inform the compressor designer of the major considerations involved in system design. It will hopefully guide him in establishing the directions for future compressor designs.

COMPRESSOR SELECTION

Once a target capacity is established, the performance of a large number of compressors can be contrasted in terms of these four major factors.

Manufacturers' data or preferably calorimeter data are collected for the operating range of the compressors (see Figure 1). A graph of condensing temperature vs. evaporating temperature for each compressor is constructed for the selected capacity. It is noteworthy that a compressor will attain the target capacity over a fairly wide range of evaporating and condensing temperatures. Due to the dependence of compressor volumetric efficiency on pressure ratio, an increase in evaporating temperature requires a corresponding increase in the condensing temperature in order to maintain a constant capacity. This effect is shown in Figure 2.

Next, lines of constant performance factor are added to the graph. Dips in the performance factor lines indicate compressors of poorer efficiency and rises indicate compressors of better efficiency. For example, compressor "B" has better performance than either of the other two compressors shown in Figure 2. The constant performance factor lines are discontinuous between compressors; however, they provide the compressor designer with an indication of the performance to be expected of a compressor displacement not shown. Note that while the same capacity can be attained with increased condensing temperatures and/or decreased evaporating temperatures as the compressor displacement is increased, this is generally accomplished at the expense of efficiency.

The slope of the compressor performance lines shown in Figure 2 are representative of presently available reciprocating compressors. Other types of compressors (e.g. rolling piston, sliding vane), may have significantly different performance characteristics. The slopes of their performance lines would vary from those shown.

As industry moves away from the traditional performance criteria in search of higher efficiency, this graph yields new design guidelines.

CONSTRAINTS ON CONDENSING AND EVAPORATING PRESSURES

Figure 2 shows that high evaporating and low condensing temperatures are indicated for maximum performance. However, practical as well as theoretical limitations exist which sharply limit the possible condensing and evaporating conditions.

The Second Law of Thermodynamics limits the evaporating temperature to the design indoor air-on temperature (usually 26°C {80°F} for residential air conditioning), and the condensing temperature to the design heat rejection temperature (usually 35°C {95°F} for air cooled equipment). Further restricting the evaporator temperature is the fact that it must remove the latent as well as the sensible load. The coil must operate wet and is therefore restricted to temperatures below the 15.5°C (60°F) dewpoint.

Heat exchangers must be of reasonable size and cost. This limits evaporating temperatures to below 10°C (50°F) and condensing temperatures to greater than 42°C (110°F). Thus it can be seen from Figure 2 that, for typical compressors, the COP is limited to about 3.5 (12 Btu/watt-hr).

SYSTEM LOSSES

System losses reduce the system performance factor from the compressor performance factors shown in Figure 2 and become more important as the performance level increases. The following items are usually encountered in systems and must be considered:

A. Refrigerant Side Losses

    1. Suction line pressure drop

    2. Discharge line pressure drop

B. Air Side Losses

    1. Condenser fan

    2. Evaporator fan or rating allowance

    3. Cabinet losses

C. Other Losses

    1. Evaporator cabinet heat gain

    2. Evaporator cabinet air leakage

    3. Control system power

REFRIGERANT SIDE LOSSES

Refrigerant side losses can easily be accounted for as shown on Figure 2. A suction line pressure drop is converted to an equivalent saturation temperature difference and subtracted from the evaporating temperature to obtain the saturated suction temperature. A discharge line loss can likewise be treated as an increase in condensing temperature.

Typically, the suction line loss is equivalent to 1°C (2°F). The discharge line loss is usually a small percentage of the discharge pressure and neglected. From Figure 2, for a selected compressor COP, unique saturated suction and discharge temperatures can be determined for each compressor.

AIR SIDE LOSSES

Air side losses are treatable in two parts: the coil pressure loss and a cabinet pressure loss. In the case of the indoor coil a reasonable amount of external pressure must also be provided to cover duct losses plus system accessories (filters, heaters, etc.).

In most air conditioning systems, air velocities seldom exceed 900 fpm and average about half that value. The dynamic component of the total pressure is small (about 0.03 cm H_2O or 0.013 in. H_2O). Significant changes in air velocity through the unit are normally avoided in order to minimize the fan power as well as the air borne sound. Thus, the dynamic pressure is both small and relatively constant. In the discussion that follows, either static or total pressure can be considered, providing the corresponding fan efficiency is employed.

The overall pressure loss through the unit is given by

$$\Delta P_{overall} = \Delta P_{coil} + \Delta P_{cabinet} + \Delta P_{external} \quad (1)$$

Further, the power consumed by the fan is given by

$$KW_{fan} = C\ \eta_f \eta_m\ CFM\ \Delta P_{overall} \quad (2)$$

where: C = constant

η_f = fan efficiency (either total or static)

η_m = motor efficiency

CFM = air flow rate

or inserting (1) into (2):

$$KW_{fan} = C\ \eta_f \eta_m\ CFM\ \Delta P_{coil} + C\ \eta_f \eta_m\ CFM\ \Delta P_{cabinet} + C\ \eta_f \eta_m\ CFM\ \Delta P_{external} \quad (3)$$

From Manufacturers' literature or actual coil tests, the coil pressure drop can be determined for the desired air flow and the first term of (3) evaluated.

The second term of (3) is of special interest.

$$KW_{cabinet} = C\ \eta_f \eta_m\ CFM\ \Delta P_{cabinet} \quad (4)$$

This term can only be evaluated from experimental data for a particular cabinet geometry. If it is assumed that the cabinet pressure loss follows a square law, i.e.,

$$\Delta P_{cabinet} \sim CFM^2 \quad (5)$$

then the cabinet losses may be found for an air flow other than the test air flow by observing:

$$KW_{cabinet} \sim CFM^3 \quad (6)$$

This is a straight line on log-log graph paper and requires only a single test point. Figure 3 shows cabinet power losses for typical coil-cabinet combinations.

If the evaporator fan is part of the system (e.g., a package), its input power must be considered part of the system power. Thus, the system is penalized (and rightfully so) for both the evaporator fan motor loss (if it is located in the air stream) and for the fan input power. If the system is for a furnace coil with no fan, 0.000447 watts/cc/sec (700 Btu/hr/1000 Cfm) per ARI rating procedure must be subtracted from the measured cooling capacity. No allowance for fan input power is made. This difference usually results in a packaged unit being penalized approximately 0.3 COP (1 Btu/watt-hr) relative to a split system.

The system designer must also be mindful of other conditions and restrictions imposed by applicable certifying agencies such as ARI (Air Conditioning, Refrigeration Institute). For example, the maximum evaporator air flow rate is restricted to 60.4 cc/sec-watt (450 Cfm/ton). The wet evaporator coil for furnace installations must not have more than 0.76 cm H_2O (0.3 in. H_2O) air pressure drop. An evaporator blower unit must be capable of a minimum amount of external static pressure at the rated air flow. It should be recognized that these requirements dictate a different optimization procedure for different systems.

Air borne noise is also a factor in design. The two primary factors are air flow (CFM) and air total pressure ($\Delta P_{overall}$).

OTHER LOSSES

Evaporator coils are commonly located in enclosures from which conditioned air may leak and heat be conducted. These losses must be accounted for in the system design. This will require an increase in compressor capacity for the same system net capacity.

Control system power is also charged to the system, however, it is generally small and neglected.

SUMMARY

There are many practical limitations imposed on components available for consideration in system design. Thus, the designer does not usually have complete freedom of choice but must often accept the discrete sizes available.

For each compressor size available, there is only one choice of saturated suction temperature and saturated condensing temperature which will produce the desired capacity and performance level. There are, however, usually several compressors that will satisfy the criteria, albeit at different combinations of saturated temperatures.

For each temperature-compressor combination, the rest of the system components must be selected. Significant variables include:

1. Coil geometry

2. Air flow rates

3. Overall package size

4. Refrigerant side losses

For any given package size, there are many combinations of coils, air flow rates and losses that will produce a workable system. The system performance, as well as cost, may vary considerably among the choices. The system designer must examine all of the possible combinations that fall within his constraints and choose the optimum one.

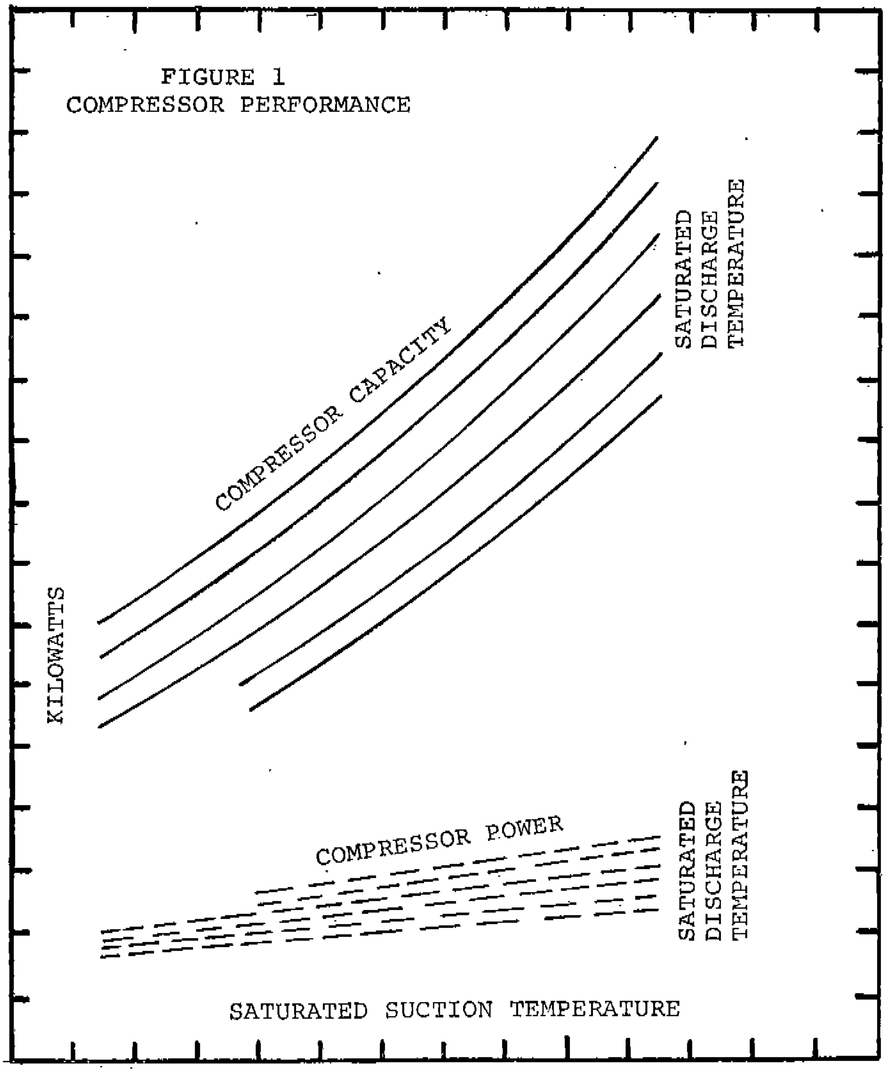
CONCLUSION

While it is true that the compressor is the "heart" of the air conditioning system, the designer must be cognizant of all of the factors which affect system performance. The compressor designer must be aware of applicable system design criteria and restrictions. He must work closely with the system designer in setting the specifications for new compressor designs.

Incomplete data, missing information and unrealistic compressor test conditions can severely hamper the efforts of the system designer. By presenting compressor information in useful forms, the interests of both the compressor designer and the system designer are served.

One final point, it is the authors' opinion that there exist many practical constraints, which limit the ultimate performance of a system. While improvements in compressor performance will undoubtedly be made, these will be second order in nature. In the short term, the "startling" increases in efficiency will come about from manufacturers reassessing the first cost-unit performance tradeoffs. Long term operating efficiency improvements will come from careful assessment of system and load requirements over the entire operating range rather than the restricted "design point" analysis favored today.

FIGURE 1
COMPRESSOR PERFORMANCE



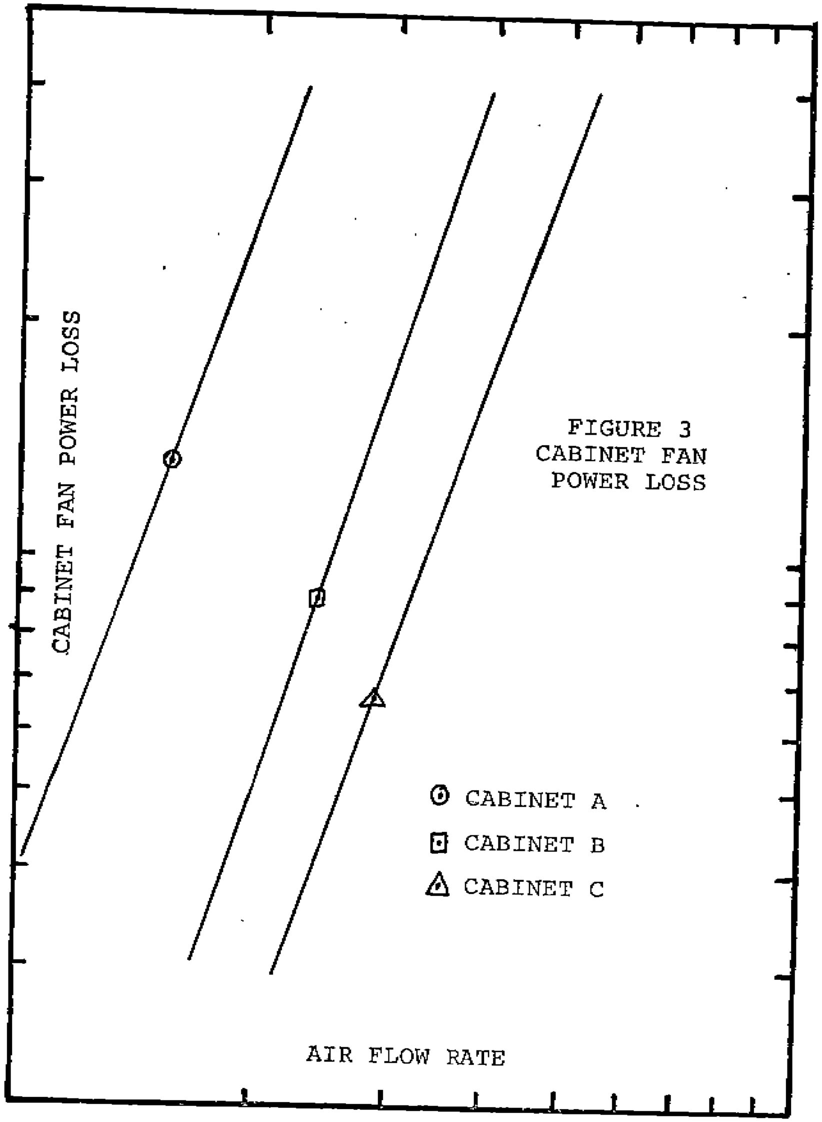


FIGURE 3
CABINET FAN
POWER LOSS

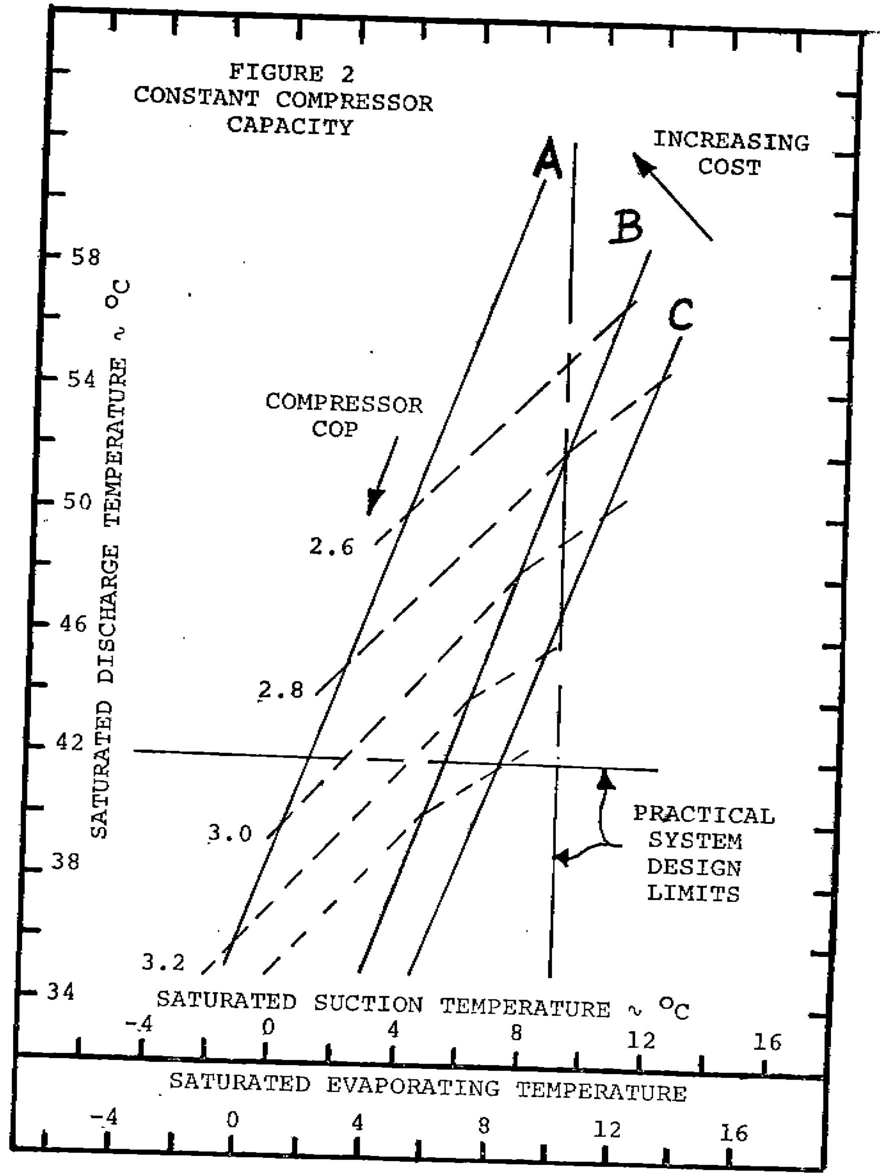


FIGURE 2
CONSTANT COMPRESSOR
CAPACITY