Design of Compressors to Meet the Performance Ranges Required for Residential Air Conditioning

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ABSTRACT

The last decade has seen an increasing awareness that the manner by which a compressor meets the part load conditions of residential air conditioning is a strong determinant of the seasonal SEER of the air conditioner. The compressor can respond to part load needs by cycling if of the single speed variety, by cycling between capacity levels if of the multispeed variety, by use of internal capacity control, or speed modulation for load following. A plot of load vs ambient temperature, showing the zero load temperatures for the class of buildings in which units containing the compressor will be applied may be used to indicate the range of loads, and indirectly the corresponding pressure ratios imposed on the compressor. Lines of capacity of such units may be included to approximate the percentage of "on" time for each ambient condition if the compressor is cycled, or the required level of capacity if the compressor is modulated. When the hours per year at the various ambient temperatures are added to the plot the expected period of operation of the compressor at each pressure ratio and average capacity are revealed. Large differences in compressor operating conditions are imposed by the air conditioning unit in different geographical areas, and by different heating-cooling changeover temperatures. The SEER of the unit the compressor serves thus varies widely in different locations. Information from such an analysis is very valuable in determining the optimum design of the compressor. Examples will be given in the paper for units located in Dallas and Chicago.

INTRODUCTION

When designing compressors for residential air conditioning applications and for heat pumps the author has for many years used a design map to establish the limits within which the compressor would need to be operated. In this approach the zero load temperatures for the buildings are estimated, while the assumption is made that the cooling design temperature is 95°F (35°C). Connecting the zero load temperature points with the design point provides an approximation of the loads and corresponding compressor capacity requirements for all ambient temperatures.

If the compressor is to be applied in a heat pump the load lines may be extended to lower ambient temperatures. The cutoff temperature for locations in which the equipment containing the compressor will be applied may be marked on the heating load line.

Additional load lines are then added to show the effect of cloudy and nighttime conditions for both heating and cooling. Increased loads will thus be indicated for heating cycles, which extend the total map area within which the compressor must operate.

The addition of lines showing the expected capacity for cooling and heat pump applications of equipment incorporating the compressor enables the determination of the cycling or modulation requirements which will be imposed on the compressor for various ambient temperatures.

The importance of any ambient on a seasonal basis depends on the relative length of time that ambient exists during the year. For many locations the average hours per year throughout the range of expected ambients are given in the ASHRAE Handbooks. For a great many other locations the hours per year at desired ambients may be calculated from the "Departure from Normal" column of the NOAA "Monthly Summary of Local Climatological Data".

The design map, so constructed, has all the information needed to estimate the SEER which various designs of compressor and of methods of meeting part loads will produce. The map is thus a powerful tool in comparing various compressor designs, and comparing capacity control methodologies.

THE BASIC DESIGN MAP

The coordinates employed in the basic map are shown in Fig 1. The practice in the United States of selecting heat pumps to meet the cooling load makes it convenient to express all loads as a percentage of the design cooling load, normally taken to occur at 95°F (35°C). This point is indicated in Fig 1. The "% Cooling Load or Capacity" scale extends to 240%, but may be extended further to any appropriate percentage desired.

The addition of the various other information for use in compressor design will be shown in successive figures for application of equipment containing the compressor to the large inventory of
Residences of the type existing prior to the 1970's. The results of a similar analysis to the type of residence which has increasingly been built since the early 1970's will then be shown for comparison, and to show the effect on compressor design.

ESTABLISHING THE LOAD PATTERN

The addition of the typical load lines for the pre-1970 residences is shown in Fig 2. These lines are representative of loads encountered in the majority of retrofit installations.

These buildings typically have zero load for sunny conditions at approximately 60°F (16°C). This is indicated by a straight line from point "D" to the zero load temperature. Experience has shown that these residences have zero load temperatures at night or during heavy cloud cover of approximately 68°F (20°C). Hence a second line may be inserted starting at this temperature for zero load and drawn parallel to the "Sunny" line in Fig 2. This line is designated "Night or Cloudy" in the Figure.

Designating the cooling design temperature, D, as 100% load as shown on the lefthand scale permits the relative load at outdoor ambient between 60°F and 95°F (35°C) to be read as a percentage of full load cooling.

The heating load lines are drawn upward to the left from the zero cooling load intersections, with a slope of 0.8 of the slope of the cooling load lines. This slope may be slightly greater for massive construction.

Fig 2 extends the heating load lines to -20°F (-29°C). Each location under consideration for application of air conditioning equipment having the compressor under consideration will have a heating design temperature as shown in Fig 2, and so marked.

Fig. 2 thus shows, by reference to its left hand scale the loads which will exist at different ambient temperatures for sunny conditions or for
nighttime and cloudy conditions, for either heating or cooling. The load for any ambient remains fairly constant wherever the building is located, and thus is building specific for a given orientation. What variations which exist as a result of location come mainly from the increased sun effects in heating or cooling in northern latitudes, which are somewhat offset by the increase in the winter solar periods in the southern latitudes.

The compressor of any air conditioning unit which is to cool the residence depicted in Fig 2 must be able to match its average capacity to the load at any ambient between the cooling design ambient as at point "D" and, if a heat pump, the "Heating Design Temperature" as shown in Fig 2. The EER or the SEER will depend on the methods used in matching the compressor capacity to the load, as by cycling, cycling between speeds, or modulating.

HOURS PER YEAR AT DIFFERENT AMBIENT TEMPERATURES

To determine the effect of compressor design on EER and SEER it is necessary to know how many hours exist at different ambients. This information is site specific. If the compressor is to be used in equipment designed for all locations in the United States one might choose contrasting cities such as Chicago and Dallas as coming close to representing the extremes of normal operation of cooling and heat pump application.

In Fig 3 the ambient hours for Chicago and Dallas have been plotted. It is best to show the hours in blocks of 5°F (2.8°C). Chicago records are shown by the longer dashes across the curves, and Dallas by the short dashes. Similar plots of hours per year vs ambient can be made for any location of interest, by reference to the ASHRAE Handbook or the Monthly Climatological Data reports mentioned above. For a general view of how a compressor will react the two contrasting cities of Chicago and Dallas are good choices, and furthermore these cities have been used in many existing comparisons. Knowing the compressor and unit performance in each block permits estimation of SEER.
Note in Fig 3 that Chicago has the maximum heating hours for a pre-1970 residence at $30^\circ F$ ($-1^\circ C$), but in Dallas such a residence would have its maximum heating hours at $50^\circ F$ ($10^\circ C$), at which condition the load is quite small. Similarly, for cooling, Chicago has a maximum number of hours at an ambient temperature of $65^\circ F$ ($18^\circ C$), while Dallas has its maximum at $78^\circ F$ ($26^\circ C$). The same residence with the same equipment and the same compressor will exhibit widely different SEER in these two locations.

Fig 3 also shows the range of ambients over which the equipment will be required to operate, or over which other means for meeting the load must be provided.

**MODES OF COMPRESSOR CONTROL**

In Fig 4 lines have been added to indicate the capacity which the compressor provides in an air conditioning unit with different extremes of compressor control. Line QC shows the unit capacity resulting from operation of a compressor without any means of capacity control. At any ambient temperature but design temperature of $95^\circ F$ ($35^\circ C$) the load requirements are met by cycling the compressor on and off. For example, at $80^\circ F$ ($27^\circ C$) ambient the load on a sunny day is shown to be 57% of design load, while the compressor operates to give the unit a capacity of 106% of design capacity. If we neglect system deterioration due to cycling, the compressor must then run $57/106 = 54\%$ of the time.

Since the unit operates at 6% above design capacity the temperature differences at the condenser and evaporator are greater than at full load, and the irreversibilities which reduce EER are also greater than at full load. For Dallas particularly this condition exists for nearly the maximum hours per year of any ambient, and thus has a major effect on SEER.

**Compressor Modulation.** The capacity of a fully modulated compressor is shown in Fig. 4 by the dash-dot line, QCM. Modulation may be obtained by such methods as intermittent suction blockage in one or
cylinders, suction valve lifters to inactivate one or more cylinders, but in either case should be done with a time constant less than 30% of the time constant of the rest of the refrigerant system. It may be necessary to include rates of change in either or both the suction pressure and discharge pressure to accomplish this rapid response, and thus produce an efficient pseudo modulation.

Complete modulation along the QCQ line may also be attained by speed control of the compressor. Thus at the 80°F (27°C) ambient both the load and the capacity of the compressor may be matched at 54% of full load. EER is then improved because the reduced compressor capacity forces a reduction in the temperature differences at the condenser and the evaporator to about 54% of their full load values. This reduction in cycle irreversibility results in a major improvement in EER, the importance of which in SEER is indicated by the relatively large number of hours per year at which this condition prevails.

Full modulation may be applied until the evaporator temperature rises to a level that prevents the required dehumidification. Control of refrigerant feed to the evaporator can accomplish this result, while the compressor capacity is permitted to continue to reduce to match load requirements and thus continue to reduce the irreversibility on the condenser side of the system.

Compressor Modulation in Heat Pumps Line QH in Fig 4 shows the capacity of a typical heat pump using the same compressor as was used in the cooling cycle. The intersections of line QH with the load lines show the "balance points", or the temperatures above which the compressor capacity has to be adjusted downward by cycling or capacity control. For ambients below the balance points auxiliary heat would be necessary to provide the difference between the capacity provided by the compressor in the unit and that required by the load lines.

The dot-dashed line, QHM, indicates how modulation can match the compressor and unit capacity to avoid
cycling. In heat pumps the advantages of compressor modulation accrue only at ambient temperatures above the balance points.

The lines QC and QH represent unit performance when the compressor is direct driven as by a 2-pole motor at a constant speed of 3450 rpm from 60 cycle current. One of the advantages of inverter driven compressors is that this limitation need not apply. By operating the compressor at higher electrical frequencies its speed may be increased to some acceptable maximum such as 6000 rpm during periods at which greater capacity would avoid the use of auxiliary heat.

The dotted line in Fig 4 shows the improvement to be expected from use of such overspeeding. During the periods of ambient temperature above the balance point modulation matches the compressor and unit capacity to the load, and at ambients below the balance temperature the need for auxiliary heat is reduced considerably. Yet this is accomplished while at the same time the optimum compressor capacity is retained during the cooling cycle. The dotted line showing the compressor and unit capacity when overspeed is used at temperatures below the balance point is designated QIM. Note also the degree to which the balance point is reduced by this overspeed.

SUPER INSULATED RESIDENCES

Fig 5 is similar to Fig 4 except that the load lines are for residences which have become increasingly popular since the fuel crises. Typically such structures are much more tightly constructed and more fully insulated. The result is that the zero load temperatures are considerably depressed. In the figure the sunny zero load point is shown as 49°F (5°C), and the night or cloudy zero load is at 55°F (12°C).

The slope of the load lines is less in these residences than in the older residences. This results in an extended cooling period, and a lower balance point at the usual compressor speeds. The range of usefulness of capacity control is thus extended.

Fig. 5, Super Insulated Residences
The area between the QIM line and the load lines is very much reduced.

If modulation is not provided in the compressor, the range over which cycling must be employed is likewise extended. The cycling losses are therefore usually greater than on the older house installation.

COMPRESSOR DESIGN FACTORS

The foregoing discussion attempts to illustrate the importance of compressor design features in determining the overall performance of the equipment in which the compressor will be used. It is shown that EER or SEER is not fixed, but is influenced by compressor design features, and by the type of residence in which equipment is installed and its location and weather features.

In several situations which the author has examined compressors were designed principally for the suction and discharge pressures and capacity needed for full load at the design point, usually a load and condensing temperature associated with 95°F (35°C). Yet Fig 3 clearly shows the different ambient temperature patterns experienced in different locations within the country. Most air conditioning equipment is expected to perform well wherever it is located. Unless conditions imposed on the compressor over the wide range of operation are input into the overall design unsatisfactory results may be experienced.

In compressors which allow leakage from the discharge into the suction side during off-periods the refrigerant may be allowed to condense in the evaporator at the termination of an operating cycle. The result is a reduction in EER by the unwanted heating of the evaporator. This is typical of some small screw compressors, and of vane type compressors which may even rotate backwards during the start of the off cycle. Small dynamic designs have been seen to have this problem.

All these factors must be thoroughly analyzed if compressor designs are to fully meet the needs of the industry to be increasingly energy efficient.

RELATED FACTORS WHICH IMPACT ENERGY USAGE

To fully utilize the capabilities of well designed compressors which incorporate modulation over a wide range, the heat transfer components and the refrigerant control of the units in which the compressors are used must be able to perform in such a way as to take full advantage of this new feature of modulation. For example, the evaporator and the condenser should be designed so that refrigerant side transfer does not materially reduce as the refrigerant velocity through the tubes decreases. Otherwise the mean temperature differences will not be proportional to load.

Refrigerant control becomes more important with the use of modulation. It is well known that a capillary tube offers optimum refrigerant control only at a few conditions of load and pressure levels. Since modulation provides capacity to match load, the capillary feed either starves the evaporator or overfeeds it under most conditions. Whatever refrigerant control is employed, it is important to avoid such a high evaporator temperature at low load that the dehumidification fails to too low a ratio. In the days when thermostatic expansion valves were the normal expansion device, models were made with maximum opening pressures to insure dehumidification.

While these related factors may not be the direct concern of the compressor designer, it is suggested that manner equipment designers depend upon the compressor manufacturer to warn of special requirements when new features are introduced.

It is hoped that the showing of the various factors covered in the figures and discussion herein will assist both in compressor design programs, and in applying the compressors which result.