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ABSTRACT

The usual residential or small commercial air conditioning systems use air cooled condensing sections. The instantaneous condensing temperature is a function of ambient air temperature, condenser air quantity, and the mass flow and superheat of the refrigerant entering the condenser. System energy requirements are currently expressed in terms of Btu/watt-hour at "design" ambient for normal applications. Present efforts within the industry seem aimed at increasing this value. Little or no regard is given to the effect on either the seasonal energy demand or on the energy required to produce the increased quantities of aluminum and copper demanded by increases in the size of heat transfer surfaces which this approach seems to employ. Thus the maximum energy effectiveness is not often met.

In this paper we will show that a different approach to energy effectiveness can give as much as 35% improvement without the need for a greater material useage in the heat transfer surfaces. Instead of using the conventional EER term of Btu/watt-hr in expressing performance, we will use here-in the inverse term, watt-hr/Btu. This permits adding the energy requirements of the various system components to determine overall system energy efficiency, and is consistent with the practice of rating large systems in terms of KW/Ton.

We provide here-in a general analysis of the effects of the interactions of ambient air temperature, the system load, and the system response for the typical "package products" unit. No analysis is made of a specific unit,- hence the results suggested may be analyzed by any product manufacturer to determine how his particular line of products can be made more energy effective on a life cycle basis by use of variable capacity compressors.

Development of such a compressor and its early use in the equipment discussed could make a greater reduction in seasonal energy requirements of residential and small commercial air conditioning equipment than any other product design change.

AMBIENT AND LOAD CYCLES

An important characteristic of the daily air conditioning load is its swing from zero, or even a negative value, during the nighttime hours to some percentage of the design load during the late afternoon. Only on a relatively few days during the normal season is the full design load reached, and then for no more than two or three hours. There are a large percentage of the days during the normal season where-in design load is never approached at all. This is the result of the diurnal temperature swing, the range of which varies from about 15° in coastal regions to as much as 35° in the dry interior regions.

Whenever the air conditioning system is called into operation the condensing temperature is mainly determined by the existing ambient temperature as shown in Fig. 1, where-in the solid line shows typical response with a conventional constant displacement compressor and an evaporator and condenser with constant air flow. Neither those systems with thermostatic expansion valves nor those with capillary control can respond over such a wide range of condensing temperature without such problems as frosted evaporators, and hence "low ambient" kits are normally supplied which force the condenser temperature along the beaded line of Fig 1.

The natural result of reducing the condensing temperature during periods when the system operates at the lower ambients would be to reduce the watt-hrs/Btu as shown by the solid line of Fig. 2. The system depicted might turn in a season average of 0.11 to 0.12 w-hr/Btu if no low ambient kit is employed, since seasonal loads tend to average about 70% of full load. The artificially high condensing temperatures imposed by low ambient kits may raise the average seasonal energy requirement to 0.14 w-hr/Btu, or even higher.

The typical residential air conditioning load varies with outdoor ambient and the level of solar insolation. It is a maximum shortly after the hot sunny period of design days, and a minimum at typical temperatures of 65°F on cloudy days, or between 50 and 55°F on sunny days. Nightime minimums of zero or of negative cooling load are reached on most days. A typical load vs ambient profile is shown in Fig. 3, which also shows by a beaded line the corresponding system capacity. Typical hours per year for the ambient temperature blocks experienced in mid-latitudes of the United States are shown in Fig. 4.

From Fig. 3 we see that where-as the load reduces sharply as the ambient decreases, the capacity of the system increases concurrently. The thermostat cycles the system to provide a percentage of "on" time which provides an average capacity equal to the load requirements. For example, Fig. 3

shows that at 75°F ambient on a sunny day the load is 60% of design, while the system capacity is 112% of design. The percent "on" time, or duty cycle, will then be 60 / 112 = 54%. Under this condition the w-hr/Btu is seen in Fig. 2 to be 0.105.

Either manually, or by computor programs, the 780 hours expected at this operating condition per year can be totalled as to energy requirement for a given system and added to the similar information from all other conditions, thus approximating the yearly energy requirement. For the system just described the annual energy efficiency may thus be shown to be 0.142 w-hr/Btu.

EFFECT OF USING VARIABLE CAPACITY COMPRESSOR

The capacity of compressors may be readily controlled even tho they operate at constant speed by any of several well known methods. Some of these methods have been described at this conference and at the 1972 Purdue Compressor Conference. In addition it is now possible to operate all electrically driven compressors at variable speed by use of invertors which control the frequency and voltage of the electrical input.

Whether displacement is controlled by mechanical devices or by speed variations, the more efficiently the compressor operates at reduced capacity the more the total energy which can be saved on a seasonal basis. Compressor input requirement at zero load equal to less than 6% of that at full capacity has been achieved.

If such a compressor with full capacity control replaces the presently used compressor in an otherwise standard air conditioning unit a modulating control system can then produce the System Capacity Line shown in Fig. 5. Note that instead of departing from the load lines at reduced load, the system capacity now tracks the load, becoming identical with it. The result is that the temperature difference between the condenser air stream and the condensing temperature becomes less and less as the load reduces. Thus the pressure ratio imposed on the compressor also reduces at the lower loads and the required w-hr/Btu does likewise.

A still further effect normally occurs at the evaporator. The reduced compressor capacity allows the evaporator pressure to rise, thus reducing the temperature difference between the air stream over the evaporator and the evaporator temperature. This still further reduces the pressure ratio and saves more energy.

MODIFICATIONS TO REFRIGERANT CONTROL

To achieve maximum gains with the system just described and at the same time to preserve the best comfort results, some changes are needed in the refrigerant control to the evaporator. A capillary cannot respond to the wide variations of pressure ratio and pressure level, nor can it vary the refrigerant flow in response to load change. Hence a thermostatic expansion valve is a better choice. However, a large reduction in compressor capacity at light loads would allow the evaporator temperature to rise to an undesireably high value, and thus to reduce or eliminate dehumidification. To prevent this undesireable action of the expansion valve-evaporator loop, it is suggested that the thermostatic expansion valve be provided with a maximum opening pressure limiting device. The pressure limit should be set to prevent the evaporator pressure from exceeding a value higher than that corresponding to 50 or 52°F.

The combination of the controlled displacement compressor and the pressure limited evaporator gives the greatest realizeable reduction in energy requirements consistent with good operation for any condenser and evaporator combination. For reductions of load to about 70% of full load the evaporator temperature is allowed to rise as the compressor capacity falls, with consequent power savings. As the load, and hence the compressor capacity, is further reduced the expansion valve limits the refrigerant flow and in effect changes the size of the evaporator to match the load at the increased evaporator temperature already attained.

SAVINGS TO BE EXPECTED

The performance of the standard system and of the system with the controlled displacement compressor is compared in Tables I thru III. It is assumed that both systems are operated automatically to maintain constant temperature in the conditioned spaces. Constant air volume is assumed for the fixed displacement compressor system, and variable air quantity for the condenser and vaporator of the system with the controlled displacement compressor. The fans are assumed to require a constant 0.010 w-hr/Btu for both systems, which means that the fans are reduced in speed to 43% of their maximum full load speed as the 60° to 65° temperature is reached in the system with the controlled displacement compressor. Such a reduction would not be possible with the standard system since it would result in higher w-hr/Btu at the full displacement at the reduced ambients.

Table I takes the hours from each block of ambient temperature from Fig 4, and applies the percent of maximum load from Fig 3 to determine the thousands of Btu's required for the season within each block. Table II determines the KW-hr needed within each block for the conventional system, while Table III determines the KW-hr for the same blocks for the capacity controlled system.

The total seasonal KW-hr required per ton of nominal full load capacity is 2006 for the conventional system compared to 1434 for the system with the capacity controlled compressor. The savings of 572 KW-hr per ton is equivalent to 29% of the power needed for the conventional system. It is achieved without the need for additional heat transfer materials or for increases in condenser or evaporator air flow. Thus the energy increases which would be required to produce added materials or increased air flow are saved. In many cases redesign of other components can make further savings in energy needs.

Other analysis, not shown here-in, indicates that the conventional system would operate 1189 hours per season, where-as the improved system will operate at its reduced average input for 3548 hours. the load seen by the utility is therefor a much more uniform load. There are fewer starts and stops, which reduces the tax on starting components and minimizes cyclical humidity swings. The lower speeds produce less noise at most loads. Furthermore the compressor could easily be started at zero or minimum capacity, thus reducing torque requirements and further simplifying the application of starting components and possibly simplifying motor designs.

This paper has analyzed only one typical system. Others which have been analyzed show energy savings ranging from 35% for the season to 28%. It is unlikely that any other single change can produce the degree of energy reduction as the wide use of capacity controlled compressors. Similar savings can be made in other air conditioning applications. In heat pumps capacity controlled compressors have raised the season COP from 2.2 to 4.2 experimentally.[1] In automotive systems even more spectacular energy savings are possible.[2]

SUMMARY

It has been shown here-in that the use of compressors with variable capacity in residential and small commercial air cooled air conditioning systems can provide a savings of from 28% to 35% of the usual energy requirements on a seasonal basis. This can be accomplished with no significant increase in the materials used, or of the physical size of present equipment. Hence no additional energy need be expended in producing the equipment having this high degree of energy savings. It is therefor suggested that an industry-wide shift to variable capacity compressors should have a high priority. The savings in energy which can be achieved by development in this direction are expected to be greater than could be made available by any other equipment design change.

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TABLE I

TYPICAL AMBIENT LOAD BLOCKS

1. Temperature Range	60/65	65/70	70/75	75/80	80/85	85/90	90/95	95/100	Total
2. Hours per Season (Fig 4)	696	782	777	582	400	220	74	17	3548
3. % Max. Load (Fig 3)	8	20	33	45	57	69	82	94	
4. 1000's Btu/Block-Ton	668	1876	3076	3142	2736	1821	613	192	14124

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TABLE II

THE EXPECTED SEASONAL PERFORMANCE OF CONVENTIONAL SYSTEMS

1. Temperature Range	60/65	65/70	70/75	75/80	80/85	85/90	90/95	95/100	Total
2. Comp. W-hr/Btu (Fig 2)	0.128	0.129	0.130	0.130	0.132	0.138	0.142	0.148	
3. System W-hr/Btu	0.138	0.139	0.140	0.140	0.142	0.148	0.152	0.158	
4. KW-hr/Block	92.2	260.8	430.6	439.9	388.5	269.5	93.2	30.3	2006

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TABLE III

EXPECTED SEASONAL PERFORMANCE
OF CAPACITY CONTROLLED SYSTEMS.

1. Temperature Range	60.65	65/70	70/75	75/80	80/85	85/90	90/95	95/100	Total
2. Comp. W-hr/Btu (2 & 5)	0.061	0.070	0.081	0.090	0.101	0.114	0.127	0.141	
3. System W-hr/Btu	0.071	0.080	0.091	0.100	0.111	0.124	0.137	0.151	
4. KW-hr/Block	47.4	150.1	279.9	314.2	303.7	225.8	84.0	29.0	1434

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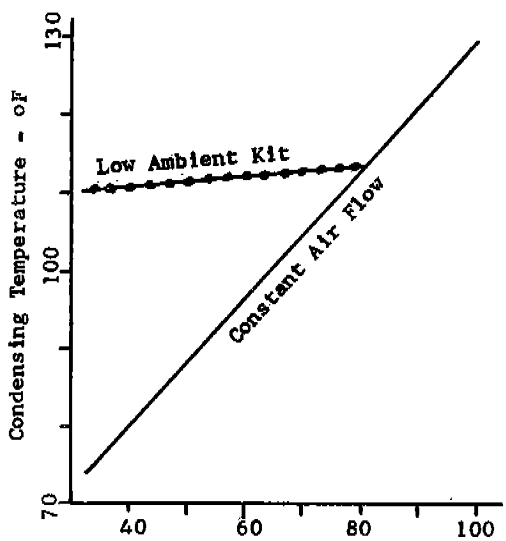

FIG. 1, CONDENSING TEMPERATURE

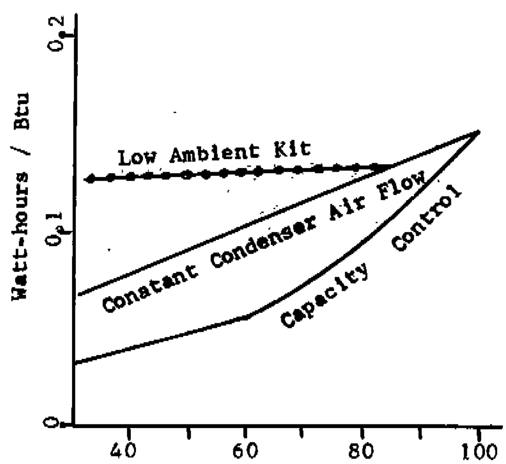

FIG. 2, ENERGY REQUIREMENTS

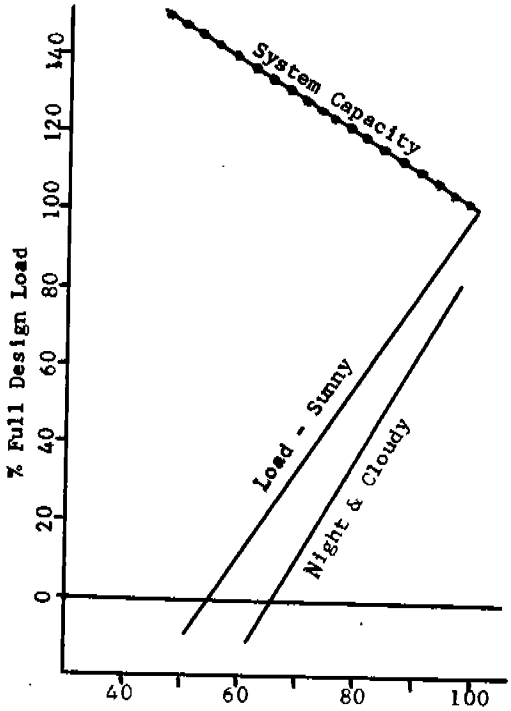

FIG. 3, LOAD vs SYSTEM CAPACITY
STANDARD SYSTEM

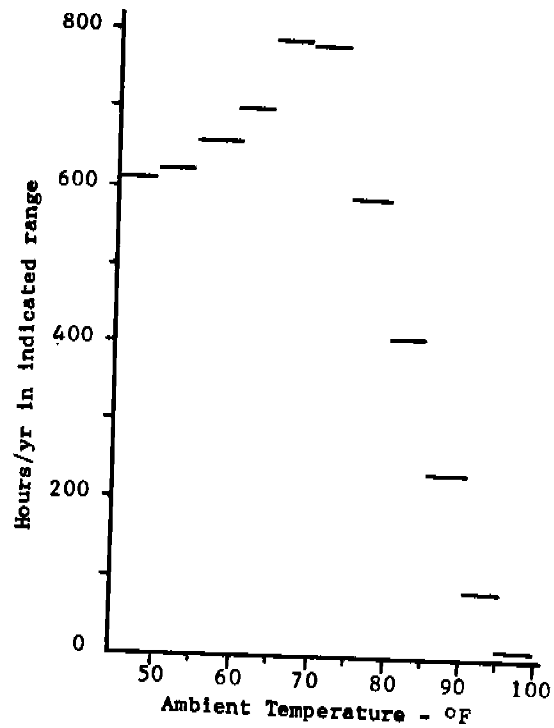

FIG. 4, SEASON HOURS vs AMBIENT

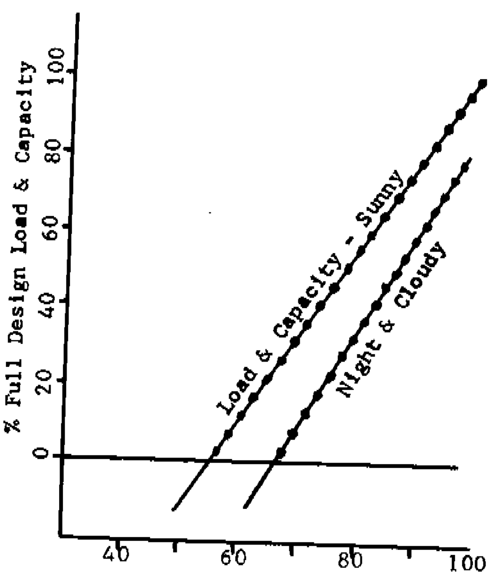

FIG. 5, LOAD vs SYSTEM CAPACITY
WITH CAPACITY CONTROL

1 Newton, A. B., "Reverse Cycle Refrigeration", U.S. Patent #2498861.

2 Newton, A.B., "How to Balance Automotive Air Conditioning Systems for Best Performance", ASHRAE Journal, Vol 15, No. 11, November 1973, Fig. 8, Page 39.