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Constant Pressure Expansion Valve Application Notes

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CONSTANT PRESSURE EXPANSION VALVE APPLICATION NOTES

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ABSTRACT

The use of constant pressure (automatic) expansion valves as the metering device in vapor compression refrigeration cycles employed in air conditioning systems offer functional advantages which result in first cost and operating cost benefits to the manufacturers and users of air conditioning equipment.

Operating performance is improved at typical application air temperature conditions and power consumption is reduced during peak summertime demand periods. This control concept should be of interest to utilities as well as manufacturers and users of air conditioning equipment.

INTRODUCTION

The constant pressure or automatic expansion valve is the fore runner of the thermostatic expansion valve and has been in production since the early days of mechanical refrigeration. It was the second stage in the evolutionary process from hand expansion valves to thermostatic expansion valves and is still employed in a wide variety of air conditioning and refrigeration applications. The recent concern for energy conservation at minimum cost has greatly expanded the interest in and demand for this simple and reliable control valve. Constant pressure expansion valves are basically pressure regulating devices which respond to the pressure at the valve outlet. Installed at an evaporator inlet, as a device to control refrigerant flow, the valve meters the refrigerant flow to maintain a constant evaporator pressure during system operation.

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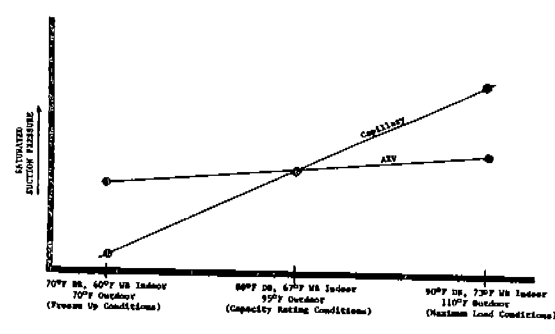
Constant Pressure (Automatic) Expansion Valves, employed as the prime expansion device, control the flow of liquid refrigerant to the system evaporator by opening on a decrease in valve outlet pressure below the set point. This results in the air conditioning system performance shown in figure 1, when tested in accordance with AHAM (Association of Home Appliance Manufacturers) standard RAC-1. For reference and comparison purposes the typical capillary tube performance is shown on the same graph. This performance data indicates that for a given system, with a constant pressure expansion valve purposely adjusted to duplicate the capillary tube performance at rating conditions, the unit will operate at higher suction pressure at low load conditions (thus achieving freeze-up protection) and will operate at lower suction pressure at maximum operating conditions (thus minimizing power consumption and protecting the compressor against overload). The example used in figures 1 through 7 is a 1 ton room air conditioner (RAC) or packaged terminal air conditioner (PTAC). Constant pressure expansion valves would have similar results when applied to any small commercial or residential central air conditioning unit through 5 ton R-22.

FIGURE 1
TYPICAL PERFORMANCE RAC OR PTAC



Through the use of system balance charts, additional benefits of employing constant pressure expansion valves on air conditioning systems can be demonstrated. These additional benefits include: improved capacity and operating efficiency through more efficient compressor operation and increased evaporator loading; and reduced first cost through condenser surface reduction and/or selection of lower capacity compressors. The potential cost reduction is more than sufficient to offset the cost of the expansion valve and increased evaporator loading. The effects of changing the evaporator pressure at rating conditions will be considered.

A complete system balance is constructed by plotting capacity performance data for all system components on one graph. A system balance plot consists of three families of curves, one representing compressor performance, another representing condenser performance and the third representing evaporator performance. All are plotted on co-ordinates of system capacity and saturated suction temperature. The family of curves representing evaporator performance can be modified by the selection of the expansion device to be employed in the system. With these curves plotted for a particular system, if any two of the following variables are known, the remaining variables can be determined: capacity, suction pressure or temperature, evaporator entering air temperature, condensing temperature or pressure, condenser entering air temperature.

FIGURE 2
SYSTEM BALANCE PLOT
COMPRESSOR CAPACITY R-22

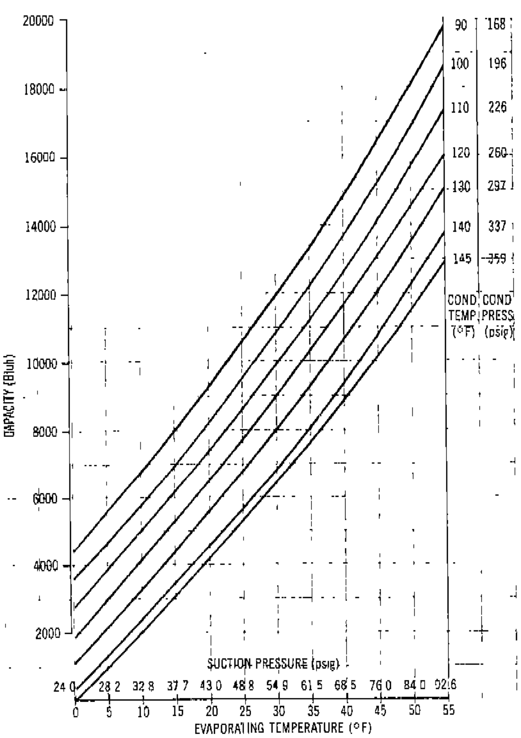


Figure 2 is a family of curves representing compressor capacity plotted against suction pressure for various condensing temperatures. Each curve indicates how compressor capacity varies with suction pressure, at a constant condensing temperature. This is typical published compressor performance data.

Figure 3 is the same compressor performance data with the addition of condenser performance data. Condenser performance (broken line) is shown as a family of curves with each curve representing a constant condenser entering air temperature. Each curve indicates condenser performance at constant condenser entering air temperature with variables of capacity, suction pressure and condensing pressure. For example: at the point where 125° condensing temperature intersects with the 40° evaporating temperature, one will note that the required condenser entering air temperature to achieve this condition is 95°. At the intersection of 45° evaporating temperature and 70° condenser entering air temperature a condensing temperature of slightly over 100° should be found.

FIGURE 3
SYSTEM BALANCE PLOT
CONDENSER — COMPRESSOR CAPACITY
R-22

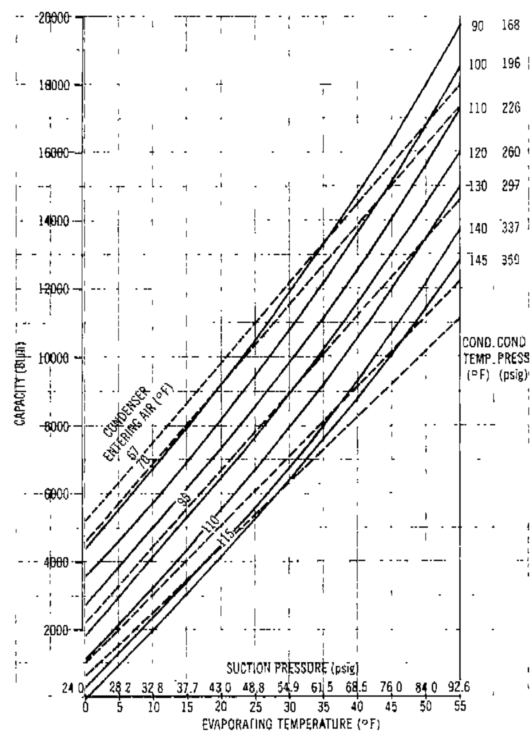
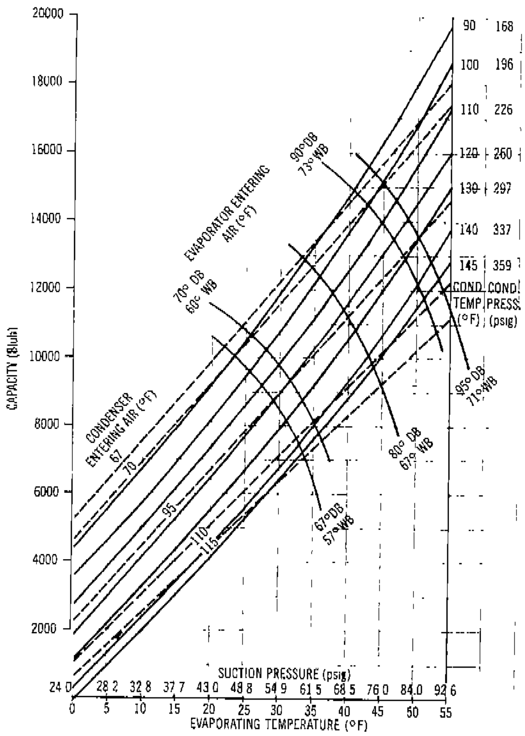


Figure 4 presents the same compressor-condenser curves with the addition of evaporator capacity performance curves for a typical evaporator using a capillary tube. Evaporator performance is plotted at constant evaporator entering air conditions. Figure 4 is a complete system balance with all three major system components represented. At AHAM rating conditions of 80° DB, 67° WB evaporator entering air temperature, and 95° condenser entering air, one will note that the system capacity is 11,200 BTU's/hr., the evaporating temperature is 40° and the condensing temperature is approximately 125°.

FIGURE 4
SYSTEM BALANCE PLOT
CONDENSER — COMPRESSER —
EVAPORATOR CAPACITY
R-22, CAPILLARY TUBE



The condenser entering air temperature and the evaporator entering air temperature were selected for these system balance charts to include both AHAM and ARI (Air-Conditioning and Refrigeration Institute) rating, low load (freeze-up) and maximum operating conditions.

In figure 4 note that at AHAM maximum operating conditions both the suction pressure and the head pressure have increased considerably over that at rating conditions. At AHAM freeze-up conditions the system represented has dropped in suction pressure to the point where freezing of the condensate on the evaporator coil will begin to occur. Note also the slope of the evaporator constant entering air temperature lines; as head pressure drops, suction pressure drops rapidly and capacity increase is limited. These observations will be useful in comparing capillary tube performance in figure 4 with constant pressure expansion valve performance in figure 5.

FIGURE 5
SYSTEM BALANCE PLOT
CONDENSER — COMPRESSOR —
EVAPORATOR CAPACITY
R-22, CONSTANT PRESSURE
EXPANSION VALVE

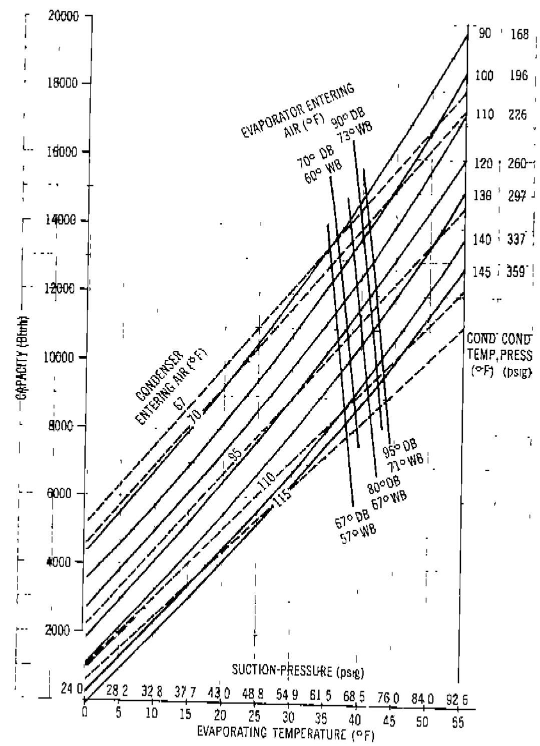


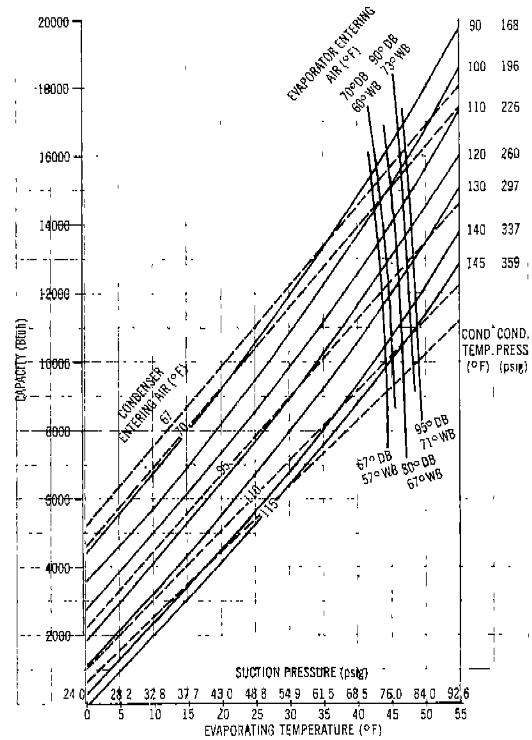
Figure 5 is a system balance plot with a constant pressure expansion valve employed rather than a capillary tube. Note that the family of curves representing evaporator performance at constant entering air temperature conditions are grouped much closer together. This occurs because the evaporator pressure is held relatively constant, within the limitations of the valve differential (typical constant pressure expansion valves require approximately 1 psi change per .001 stroke). Some additional separation of the low load condition curves from the rating condition curve is due to a normal tendency at low evaporator load, in constant pressure expansion valve applications, for liquid refrigerant to collect in the evaporator rather than the condenser. This results in flash gas in the system liquid line which in turn requires additional valve stroke to maintain the pressure level desired to prevent freeze-up. The slope of the evaporator entering air temperature curves is greater for the constant pressure expansion valve performance due to the valve's ability to open as suction pressure tends to decrease in response to decreasing head pressure, thus maintaining relatively stable evaporator pressure and temperature. This results in higher system capacity for low air temperatures entering the condenser. Note that the point representing AHAM maximum operating conditions occurs at both a lower suction pressure and a lower head pressure than in the case of the unit employing a capillary tube. This results in a reduced power consumption level at high loads, when the demand on electric utilities for power is greatest and when the capillary tube unit consumes the greatest amount of power. This energy conservation is achieved at only a slight loss in capacity at maximim operating conditions when compared to the ARI rating condition or to the capillary tube unit.

The improved capacity, as head pressure drops, for a constant evaporator entering air temperature condition is a significant advantage for units operating in buildings with the possibility of relatively high indoor loads combined with low outdoor ambient. This can occur in installations with high loads due to people, lighting or equipment operating in a conditioned space. Also, in installations where the building is unoccupied during daylight hours. Here the entire structure is soaked in high temperature and sunlight during the day, and the air conditioning unit is not turned on until the evening hours when the outdoor ambient has dropped.

At AHAM low load conditions in figure 5 both the suction pressure and head pressure are higher than the corresponding point for capillary tube operation in figure 4. Graphically then, this point represents the freeze-up protection which constant pressure expansion valves are capable of providing by maintaining the evaporator pressure at a high level regardless of the head pressure and evaporator entering air temperatures. While the freeze-up protection is accomplished by over-feeding the evaporator, this over-feeding is much less severe than that which occurs when the evaporator on a capillary tube unit freezes up. Through careful selection of valve orifice size the over-feed can be held to the minimum required to just prevent freeze up at the minimum air temperature conditions required by the customer. In effect the slight overfeed which occurs during operation at low load, for freeze-up protection, is protection against severe flood-back which could occur if the evaporator were allowed to freeze-up. Liquid overfeed reaching the compressor can be minimized by avoiding free draining evaporator designs, an accumulator is not required.

Figure 6 represents the same unit with a slightly larger evaporator and/or increased evaporator air flow rate. The constant pressure expansion valve is adjusted to a higher pressure level at the AHAM rating condition. At rating conditions the evaporator temperature is now 46°, the capacity 12,600 BTU/hr. and the condensing temperature 128°. The compares to 40°, 11,200 BTU/hr. and 125° respectively at

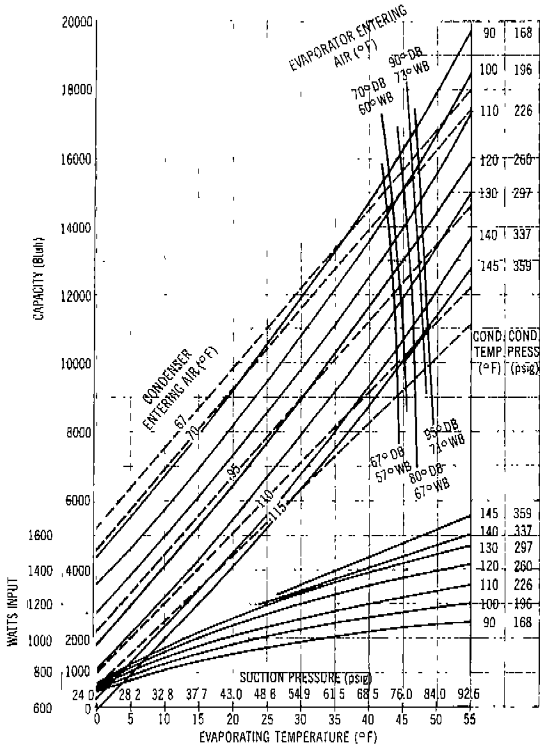
FIGURE 6
SYSTEM BALANCE PLOT
CONDENSER -- COMPRESSOR --
EVAPORATOR CAPACITY
R-22, CONSTANT PRESSURE
EXPANSION VALVE



rating conditions for the capillary tube unit and, is achieved without any increase in head pressure at maximum operation conditions. Thus the unit rating could be upgraded with only the addition of evaporator surface and/or evaporator air flow rate. Of course, the manufacturer could also elect to reduce the compressor and/or condenser size to keep the system capacity at the original level while reducing the factory cost of the unit through the use of a smaller compressor and less condenser surface. The condenser surface reduction could be accomplished several ways: reducing the number of fins per inch, reducing the tubes in the face, or reducing the diameter of the tubes in the condenser for example from 3/8 to 5/16. Reducing the number of tubes or tube diameter (Prime surface) will result in additional savings thru reduced refrigerant charge. Freeze-up protection of course is still provided thus eliminating the need for cross ambient type thermostats and allowing the use of a low cost bimetal inbuilt thermostat.

Figure 7 is a typical system balance plot (same as figure 6) with the compressor watts input included as a separate family of curves on the lower portion of the diagram. Using this plot the BTU/watt hr. ratio was determined for the AHAM freeze-up, rating and maxi-

FIGURE 7
SYSTEM BALANCE PLOT
CONDENSER — COMPRESSOR —
EVAPORATOR CAPACITY
R-22, CONSTANT PRESSURE
EXPANSION VALVE



mum operating conditions and a comparison made between the capillary tube unit and the automatic expansion valve unit with the following results, indicating that the automatic expansion valve application is operating on a higher efficiency portion of the compressor performance curve at rating and freeze-up conditions due to the higher suction pressure:

EER - (BTU/WATT-HR)

AHAM	RATE	FREEZE	MAX
AXV	8.87	12.92	6.69
CAP. TUBE	8.24	10.92	6.76

While operating slightly less efficiently at maximum operating conditions the power consumption, in absolute terms is lower for the AXV unit (1600 watts vs. 1700 watts).

Additionally, a system employing a constant pressure expansion valve is not a critically charged system. The valve will compensate automatically for over and under charge situations.

Temperature and humidity control is improved and the stress on starting components is reduced by the elimination of thermostat cycling for freeze-up protection.

The off-cycle unloading feature is usually provided in the valve to allow the use of the same low starting torque compressors employed with capillary tubes.

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