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COMPRESSOR TECHNOLOGIES FOR
R-22 LOW TEMPERATURE APPLICATIONS

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ABSTRACT

Substitutes for commonly used refrigerants CFC-12 and CFC-502 are not commercially available. HCFC-22 is not suitable for low temperature applications because of the high discharge temperature caused by high compression ratios. Staged compression and liquid injection are two approaches to prevent compressor overheating in HCFC-22 low temperature applications.

This paper describes the above approaches and their efficiency as compared to conventional applications.

INTRODUCTION

Recently, there has been much debate on the effect of CFCs on the depletion of the ozone and the global warming caused by the greenhouse effect. This issue is forcing the refrigeration industry to change directions in search of new safe substitutes and alternatives to replace CFC-12 and CFC-502.

The refrigerant manufacturing companies are in the process of evaluating direct replacements such as HFC-134a and HFC-125. However, these refrigerants are still in the development stages and might be a few years before they are available in large quantities that could satisfy the demand of the industry.

One alternative available today is to use HCFC-22 as an interim substitute. HCFC-22 is available in large quantities, has a very low ozone depletion factor [1], and has a greenhouse potential factor that is comparable to the new proposed refrigerants.

HCFC-22 could easily be used for high and medium temperature applications. However, there is some difficulty in using it as a low temperature refrigerant. In low temperature applications (-40°F to 0°F), HCFC-22 operates with high discharge temperatures as shown in Figure 1. The discharge temperature of the gas leaving the cylinders increases to a point where it exceeds the operating limit of the refrigeration oil and may damage or shorten the life expectancy of the compressor, particularly under high compression ratios and continuous operation. To assure long compressor life and minimize compressor failure, the discharge temperature of the compressed gas in the cylinders should not exceed about 300°F.

There are several techniques that could be utilized to control the discharge temperature and prevent compressor overheating. The most practical of these methods are discussed below.

LIQUID INJECTION AND DEMAND COOLING

Liquid injection techniques have long been used in refrigeration systems in an effort to limit or control excessive discharge gas temperatures. Most of the prior methods utilized capillary tubes or thermal expansion valves to control the fluid injection. These systems have been inefficient, have flooded the compressor, and have decreased the reliability of the compressor.

The demand cooling method developed by Copeland Corporation [2] overcomes these problems by providing a liquid injection technology which utilizes a temperature sensor positioned within the discharge chamber of the compressor in close proximity to and in direct contact with the compressed gas exiting the compression chamber. Thus, a more accurate indication of the compressor heating is achieved.

An electric solenoid valve coupled with a pre-selected orifice controls the flow of the refrigerant from the condenser to the suction chamber of the compressor as close to the cylinders as possible. The injection location is selected carefully to assure even refrigerant flow to each cylinder to insure maximum efficiency and even cooling effect.

The temperature sensor constantly senses the discharge temperature and provides a signal to an electronic controller which cycles the valve, allowing the required amount of two-phase mixture to be injected to the compressor as presented in Figure 2.

Mathematical modeling of semi-hermetic compressors with $65^{o}F$ return gas temperature determined the minimum amount of refrigerant that must be bypassed back to the suction chamber which will prevent the discharge gas from overheating and protect the compressor from flooding. Figure 3 presents a graph that relates the injection amount as a ratio of the total mass flow of the compressor.

The orifice in the valve is sized for each compressor to provide maximum fluid flow at a pressure differential of about 300 psi which approximately corresponds to an evaporating temperature of $-40^{o}F$ and a condensing temperature of about $130^{o}F$. This represents a worst case design criteria. It is important that the orifice be sized to create a pressure drop which is equal to the pressure drop occurring between the condenser outlet pressure and the compressor suction pressure to prevent the evaporator from backing up.

STAGING

An effective way of decreasing the discharge temperature of an HCFC-22 low temperature application is by reducing the compression ratio through the compressor. This reduction can be achieved by staging the compression process in two separate steps as shown in Figure 4. The flow from the evaporator enters the low stage cylinders and its pressure is raised from suction pressure to an intermediate pressure. The vapor runs through a cooler to reduce its temperature. The gas then enters the cylinders of the high stage where the pressure is raised to the condenser pressure.

The intermediate pressure is defined as the pressure at which the gas leaving the low stage enters the high stage. Theoretically, the [3] performance of a staged compressor is maximized when the interstage pressure (Po) equals the geometric mean of the low stage suction (PsL) and the high stage discharge (PdL) pressures.

$$Po = \sqrt{PsL * PdH}$$

The graph shown in Figure 5 presents the optimum interstage pressure calculated at different evaporating and condensing temperatures. The interstage pressure is controlled by the volume flow rate of the stages. Increasing the volume flow through the high stage or decreasing the volume flow through the low stage drops the interstage pressure.

Interstage cooling between stages must be considered very carefully with respect to the initial component's cost, system simplicity, and overall performance.

Several methods could be used. The simplest is liquid injection between stages where liquid refrigerant from the condenser is expanded and mixed with the discharge gas of the low stage in a heat exchanger or an accumulator. The amount to be injected must be controlled to cool the gas entering the high stage at 65°F or whatever return temperature is recommended by the compressor manufacturer. In some applications where plenty of cold water is available and the intermediate pressure is high, it is desirable to use a water cooled intercooler or a combination of liquid-injection and water cooling. In more complex systems, a flash tank and two expansion valves could also be used. Flash tanks are expensive, large in size, and require special valves and sizing techniques in order to operate correctly. Other cooling techniques such as external heat removal via a heat exchanger could provide significant improvement on efficiency as will be discussed later.

INTERNALLY COMPOUNDED COMPRESSORS

Many applications require the use of one single compressor. Therefore, the staging process must be accomplished in the same machine. The cylinders inside the compressor are internally compounded into two separate stages.

An analytical model was developed to calculate the gas volume flow relations between the low stage and the high stage operating at different conditions. In the model, the interstage pressure was optimized by using the geometric mean equation. The calculated results are presented in Figure 6. Usually, the low to high stage cylinder ratio is selected to be 2:1, which optimizes performance at some points and decreases it at the rest of the compressors envelope.

Practical experience with internally compounded compressors made by different manufacturers indicated that the actual intermediate pressure is 10 to 18 PSIA lower than the optimum pressure calculated using the geometric mean relation. Internally compounded compressors have the tendency to operate with higher efficiency than conventional single stage compressors only when they are operated at evaporating temperatures lower than -35°F and condensing temperatures higher than 120°F.

Internally compounded machines present a solution to the high discharge temperature problem when HCFC-22 is used. However, this technology is expensive, bulky, and can cause excessive vibration which decreases the reliability of the system.

EXTERNALLY COMPOUNDED COMPRESSORS

In a typical supermarket application, many compressors are installed in parallel on a rack to provide low temperature refrigeration for several freezer cases. At the present time the majority of these units are single staged CFC-502 compressors (multiplexes).

A way of decreasing the discharge temperature of a HCFC-22 low temperature rack is by reducing the compression ratio through the compressor by rearranging the plumbing of the compressors and compound them externally as shown in Figure 7. With this technology, the flow from the evaporator enters the low stage compressors. The discharge of the low stage then is cooled as described previously. Finally, the gas enters the high stage compressors where it is compressed to the condensing pressure.

The gas volume flow relations with this type of staging is similar to those presented in Figure 6. The advantages of this method over internally compounded compressors is the ability to modulate both the low stage and high stage to match the load requirements and also for optimizing the intermediate pressure, therefore optimizing the efficiency of the compressors. Capacity modulation could be accomplished by many different methods. The most common are variable speed, blocked suction, and variable re-expansion methods.

With externally compounded compressors, lubricant oil has the tendency to be trapped in the intercooler or in one of the stages. The oil level must be watched and corrected in each of the low and high stage compressors. To manage [4] the oil distribution in a two stage externally compounded system, an oil separator is usually installed between the high stage compressors and the condenser. The oil then is returned to the high stage crankcases. Return lines and float valves connect to the crankcases of both stages to constantly equalize the oil levels in all compressors.

In sizing the compressors for an externally compounded system, the low stage capacity is sized in the same manner as a single compressor. The compressors are selected to provide the capacity required at the evaporator temperature and the saturated temperature equivalent to the intermediate pressure. It is important to remember that the low stage compressor data is usually published at some liquid line temperature; therefore, temperature correction factor must be used to correct for capacity.

High stage compressors are chosen to handle the low stage load, the power input to the low stage, and any other additional loads if applied which could be medium temperature load, interstage cooling, and the load used in subcooling the low stage liquid. It is always a good practice to consider the following when sizing an externally compounded system:

1. Allow 3^{o}F drop between the low stage compressors and the intercooler.

2. Allow 3^{o}F drop between the intercooler and the high stage compressors.

3. Allow 3^{o}F drop between the high stage compressors and the condenser coil.

4. Oversize the low stage by 10 to 15%.

BOOSTER SYSTEMS

In large supermarket stores where there is a need for low and medium temperature refrigeration, the racks in the store could be compounded together where booster compressors are used to increase the low evaporating pressure to the suction pressure of the medium rack as shown in Figure 8.

In such a system, no cooling is required between the two stages if the medium temperature rack is operating at 25^{o}F evaporating or lower. The temperature of the gas mixture entering the high stage suction manifold depends on the mass flow rate ratio between the stages. The graph shown in Figure 9 presents the maximum flow rate ratios between stages that prevent the discharge temperature of the high stage from overheating above 293^{o}F.

SEASONAL EFFICIENCY (SEER)

In this section, energy consumption of the previously described methods is analyzed. The analyses were completed for single compressor systems and multi-compressor systems. The results of the study are presented as a ratio of the total heat removed in BTU during a period of twelve months to the total energy input in watt-hrs during the same period. The analyses were conducted for typical applications in the U.S.A. The following assumptions were considered in these analyses.

1. The available datum for all compressors is correct.

2. St. Louis, Missouri, weather data.

3. 112°F design condensing temperature, 70°F minimum condensing.

4. Degradation coefficient of 0.4 for single compressor systems.

5. Heat load of 170 KBtu/hr. for all multi-compressor systems and 38KBtu/hr. for all single compressor applications.

6. 65°F return temperature to all compressors.

The results of the analyses are presented in Figures 11 and 12. For single compressor applications, HCFC-22 systems are up to approximately 10% less efficient than conventional CFC-502 systems. However, for multi-compressor applications, externally compounded systems are more efficient than conventional CFC-502 systems. In some cases, the efficiency could be 10 to 15% higher than CFC-502 single stage system. The overall efficiency depends on the hardware used, the control process, and the operating conditions.

RECOMMENDATIONS AND CONCLUSIONS

Many different techniques could be utilized for operating an HCFC-22 low temperature system. Each of these techniques presents a solution to the high discharge temperature problem. However, to optimize performance and reliability, the following is recommended:

1. For single compressor applications down to -40°F evaporating temperature, demand cooling technology presents an ideal solution.

2. For multi-compressor rack applications, externally compounded staged compressors is the logical approach.

3. In applications where staging is not possible, parallel demand cooled compressors could be multiplexed together on a rack.

REFERENCES

1. J. R. Sand, E. A. Vineyard, V. D. Baxter. Alternative Refrigerants For Vapor Compression Refrigeration/Air Conditioning Applications. IEA News Letter, Vol. 7, No. 2, June 89.

2. Tariq A.R. Diab. Compressor System With Demand Cooling, Patent Pending Approval, 1989.

3. W. F. Stoeker. Refrigeration and Air Conditioning book, McGraw-Hill Book Company, 1958.

4. Guy R. King. Basic Refrigeration book, Business News Publishing Company, 1986

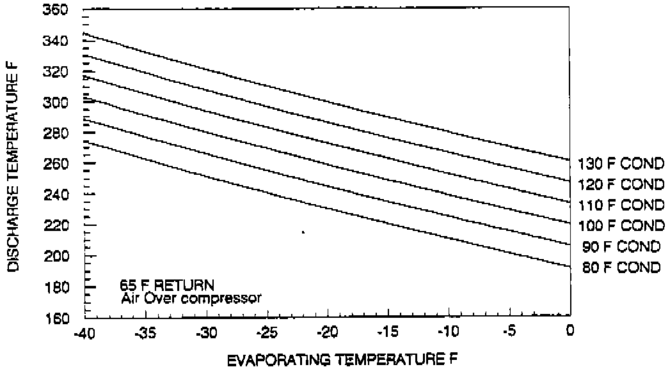


Figure 1 - MINIMUM DISCHARGE TEMPERATURE

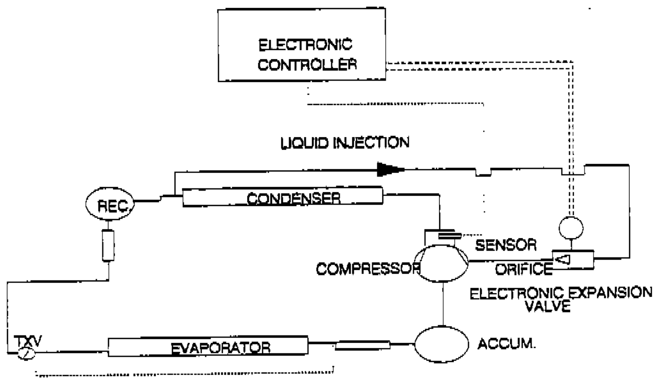


Figure 2 - DEMAND COOLING METHOD

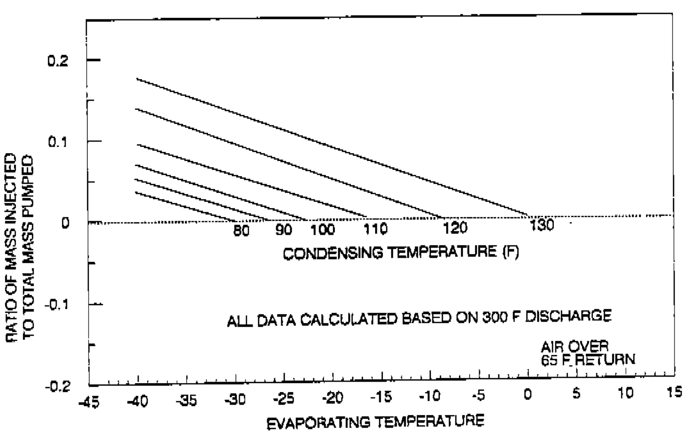


Figure 3 - INJECTION MASS FLOW RATIO

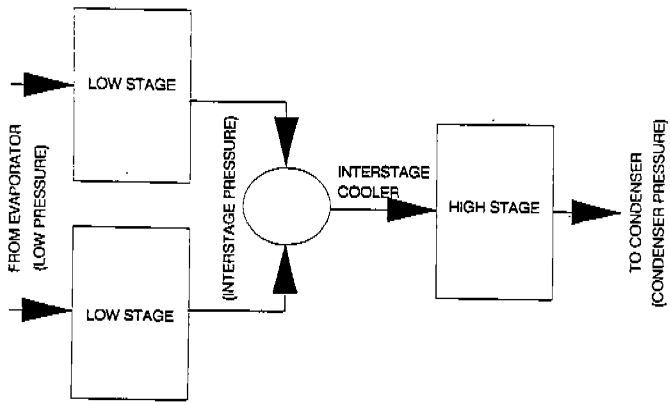


Figure 4 - STAGED COMPRESSOR

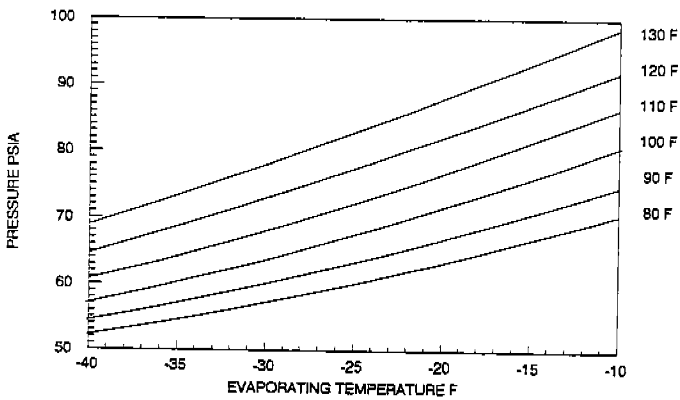


Figure 5 - OPTIMUM INTERSTAGE PRESSURE

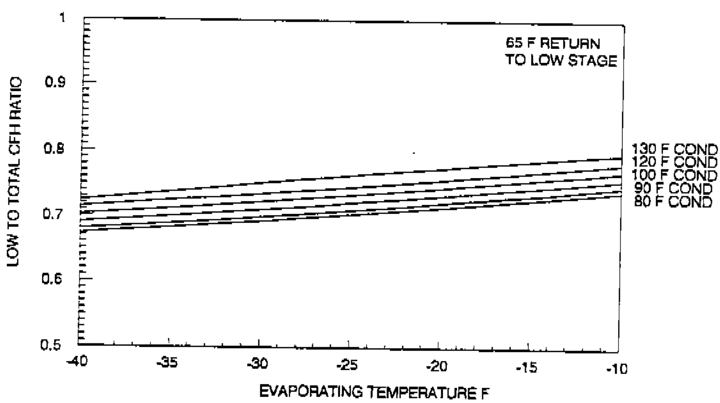


Figure 6 - FLOW RATIO OF THE LOW STAGE TO THE TOTAL FLOW

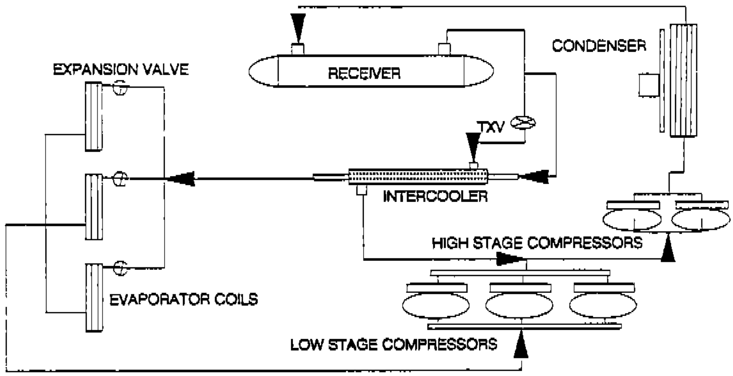


Figure 7 - EXTERNALLY COMPOUNDED SYSTEM

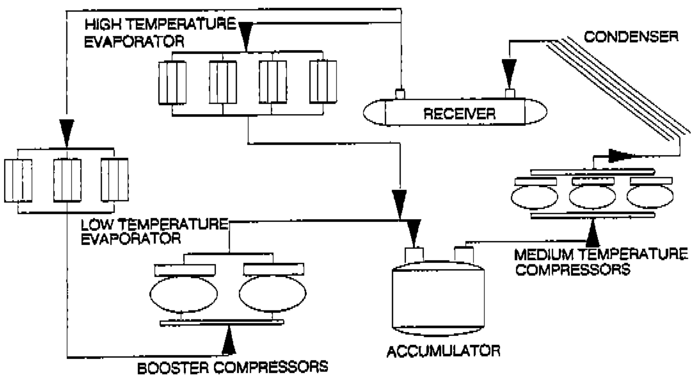


Figure 8 - BOOSTER COMPRESSOR SYSTEM

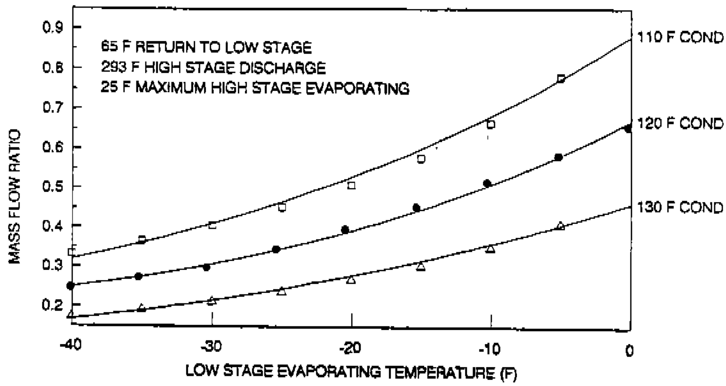


Figure 9 - MAXIMUM FLOW RATE RATIO
(BOOSTER COMPRESSORS TO MEDIUM TEMPERATURE COMPRESSORS)

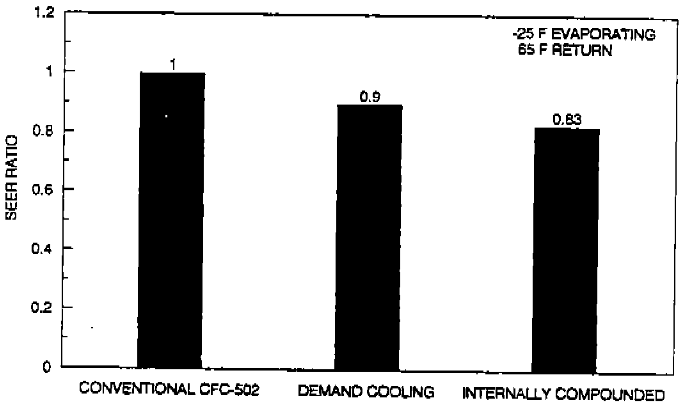
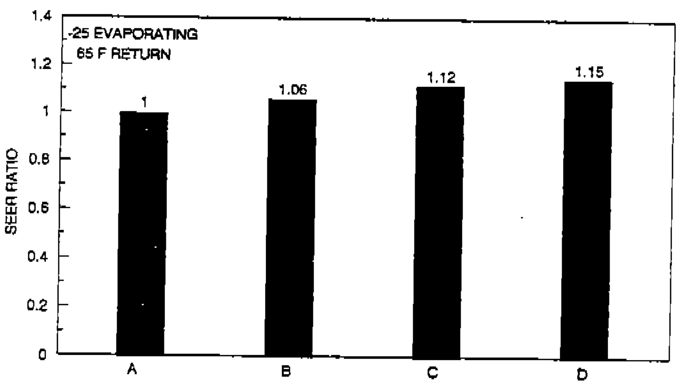


Figure 10 - SEER RATIO FOR SINGLE COMPRESSOR SYSTEM



A= CONVENTIONAL CFC-502 SYSTEM
B= EXTERNALLY COMPOUNDED WITH LIQUID INJECTION
C= EXTERNALLY COMPOUNDED+LIQUID INJECTION+15 F SUBCOOLING
D= EXTERNALLY COMPOUNDED WITH EXTERNAL HEAT REMOVAL

Figure 11 - SEER RATIO FOR MULTI-COMPRESSOR SYSTEM