T37 Development of a Variable Volume Ratio Screw Compressor

International Institute of
Ammonia Refrigeration
1200 19th Street, N.W.
Washington, D.C. 20036-2401

DEVELOPMENT OF A VARIABLE
VOLUME RATIO SCREW COMPRESSOR

Prepared for

IIAR Annual Meeting
April 17 - 20, 1983

Prepared by:

J. W. Pillis
Project Engineer
Frick Company

MARCH 3, 1983

INTRODUCTION

The double helical rotary screw compressor has been successfully applied in the refrigeration industry since the mid to late 1950's.

Characterized by high volumetric and adiabatic efficiencies, stepless capacity control, ability to operate over a wide range of compression ratios, pulsation free operation, and good reliability, the screw compressor has gained a significant position in the refrigeration market. However, one factor that has limited the field of application of the screw compressor has been its "fixed volume ratio". The introduction of the "movable slide stop" has eliminated this restriction by allowing the compressor's volume ratio to be adjusted during operation. This innovation allows significantly improved efficiency over wide ranges of application.

In order to explain how the moveable slide stop works, the basics of screw compressors will be briefly reviewed.

PRINCIPLES OF OPERATION

Compression of gas in a screw compressor is caused by the rotation of the male and female rotors trapping gas in a pocket between them. The available volume in the trapped pocket is reduced as the two rotors mesh, carrying the gas charge toward the discharge end of the machine while increasing its pressure. This can be seen schematically in Figure 1. The compression will continue until some relief is provided. Generally, discharge ports are located both radially and axially at the discharge end of the rotors to allow the trapped gas to

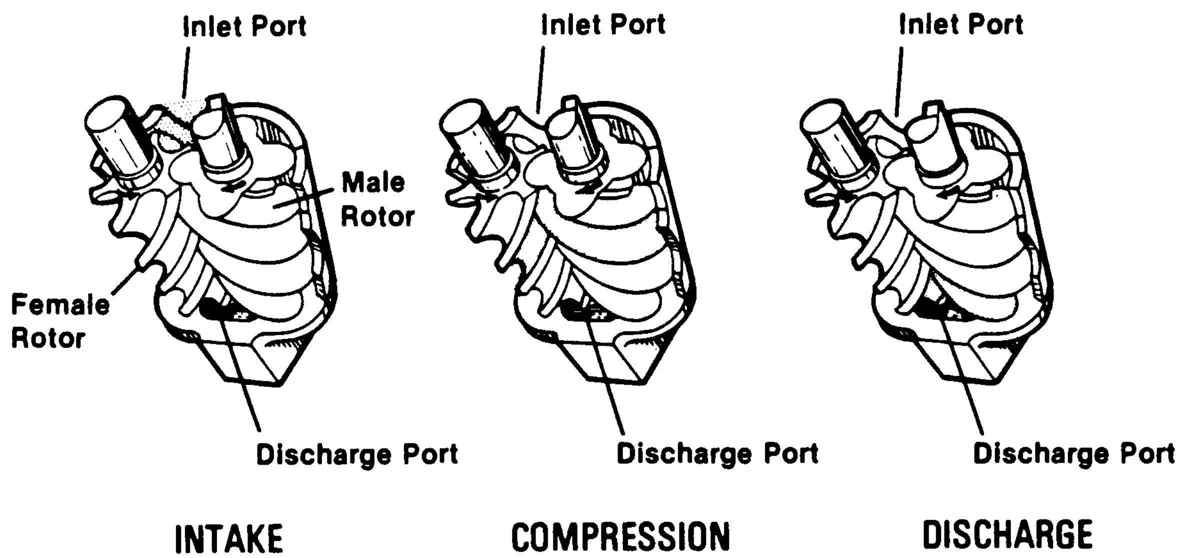
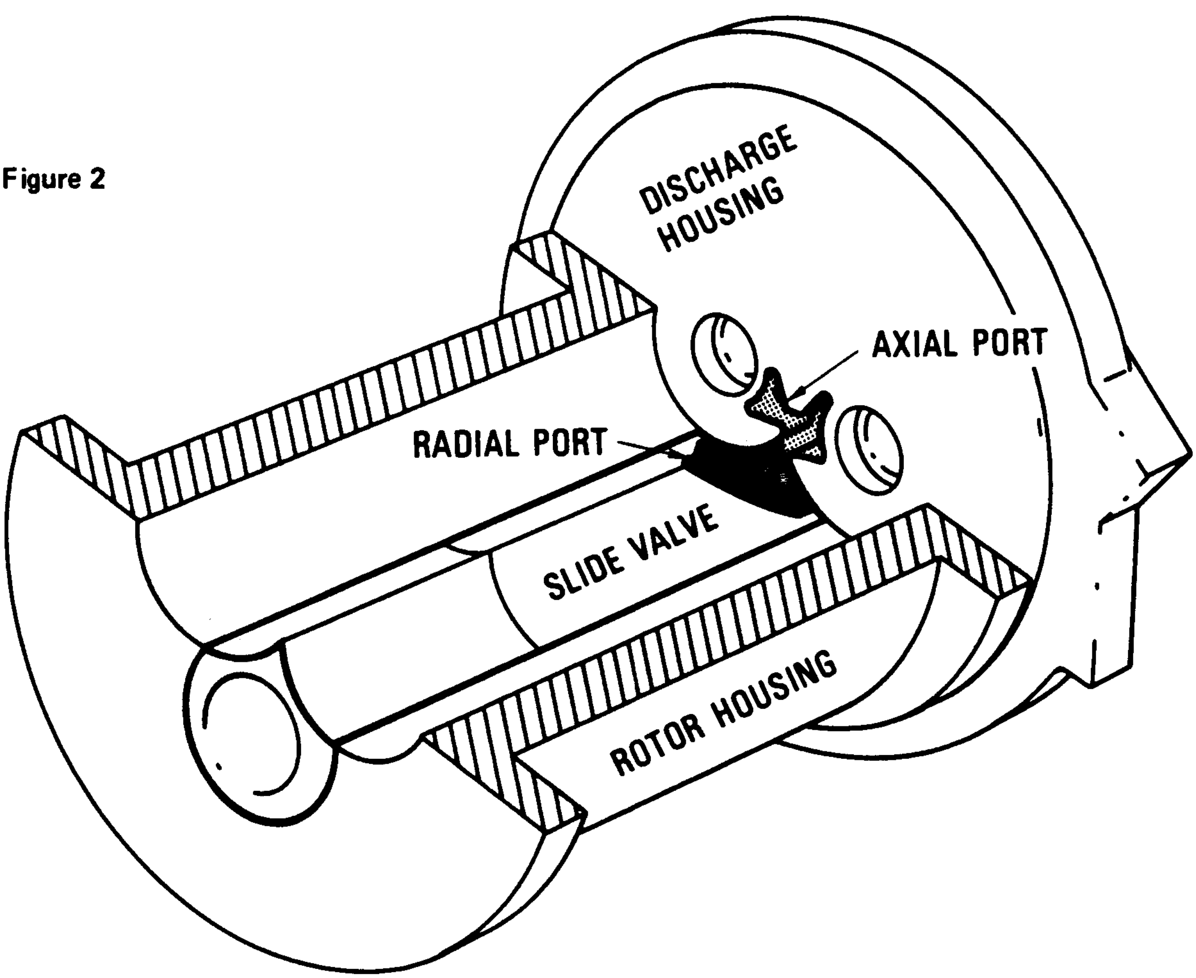


Figure 1

Figure 2



escape (See Figure 2). Once the trapped pocket opens to the discharge port, compression ceases (ideally) and the gas charge is transferred into the discharge cavity.

The amount of internal compression which occurs in a particular machine prior to discharge, and hence the discharge pressure, is therefore a characteristic which can be controlled by the location of the discharge ports. At the time of manufacture, the ports are located to yield a machine with a fixed, ("built in") pressure ratio (Pi).

$$Pi = \frac{Pd}{Ps} \qquad \frac{\text{Internal pressure in the trapped pocket immediately before it opens to the discharge port}}{\text{Internal pressure in the trapped pocket immediately after it closes to the suction port.}}$$

The term "built in volume ratio" is more frequently used than built in pressure ratio, and is defined as follows:

$$Vi = \frac{Vs}{Vd} \qquad \frac{\text{Volume of the trapped gas pocket immediately after it closes to suction}}{\text{Volume of the trapped gas pocket immediately before it opens to discharge}}$$

The built in volume ratio is related to the internal pressure ratio by the following formula:

$$Pi = Vi^K \quad \text{Where } K = \text{ratio of specific heats for a given refrigerant}$$

The importance of these concepts becomes evident when applying a fixed volume ratio compressor to a given refrigeration system. The pressure in the refrigerant condenser (system discharge presssure) is controlled only by the condensing temperature, while the discharge pressure from the compressor is controlled only by built in volume ratio and the suction pressure.

Inspection of Figure 3 shows four pressures of concern. Suction pressure of the system and the internal suction pressure in the compressor will always be approximately equal, except for slight pumping losses. However, the internal discharge pressure from the compressor can be either above or below system discharge pressure. Either condition causes efficiency losses.

If the compressor's Vi is too high for the system compression ratio over-compression losses occur. The trapped gas is compressed to a pressure above condenser pressure and then expands into the discharge line when the port is uncovered. The work done in compressing to the higher pressure is lost.

If the compressor's Vi is too low under-compression losses result. These losses occur when the compressed gas opens to a higher discharge pressure outside the closed thread. The compressor must then pump against this higher pressure for the remainder of the thread mesh, requiring higher torque than would have been required if the discharge port were located later in the compression.

Efficiency losses resulting from unmatched compression are evident in compressor performance data. Figure 4 shows typical adiabatic efficiency curves for four screw com-

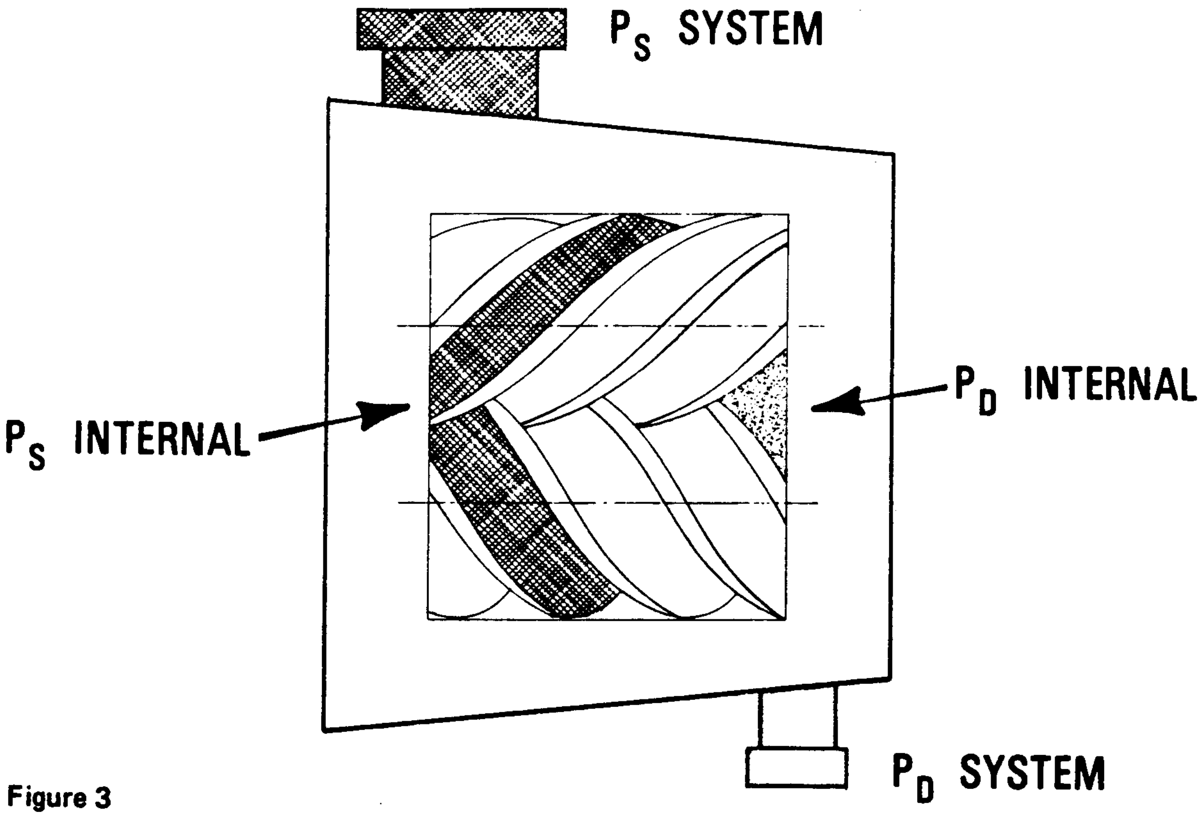


Figure 3

SCREW COMPRESSORS

EFFICIENCIES with FIXED VOLUME RATIO

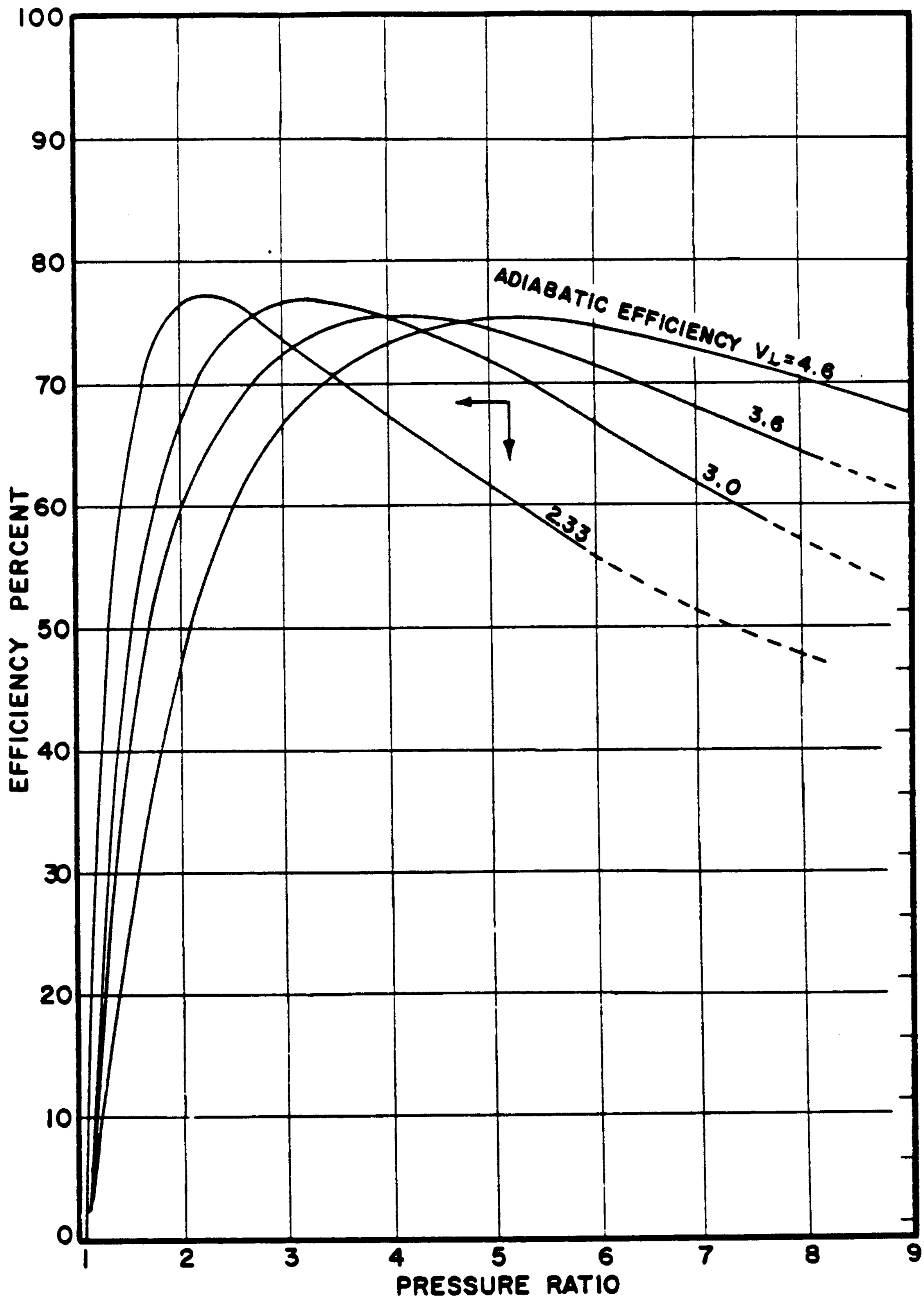


Figure 4

pressors of different fixed Vi's.

Note that a given compressor must be operated in a very narrow band of system compression ratios or considerable losses can occur.

For this reason, in compressor selection, the Vi is of major concern and every effort is taken to specify a compressor which will be optimum at the expected system compression ratio. As long as the system operates at the expected ratio, ideal compression will occur. However, many systems will operate at different evaporation and condensing pressures during different times of the year, or depending on the type of product being cooled or frozen at different times. Ice making equipment will often see different conditions depending on whether ice is being formed or harvested.

Application of fixed Vi compressors to such systems obviously can add to user energy costs.

This problem has been recognized for many years and numerous methods have been proposed for changing a screw compressor's volume ratio. However, only since the world has become more energy conscious, and microprocessor controllers have become cost competitive, has the variable volume ratio screw compressor become viable.

THE MOVEABLE SLIDE STOP

Slide valves have been employed for many years to allow screw compressors to operate at reduced capacity. These operate by opening a recirculation passage to by-pass a portion

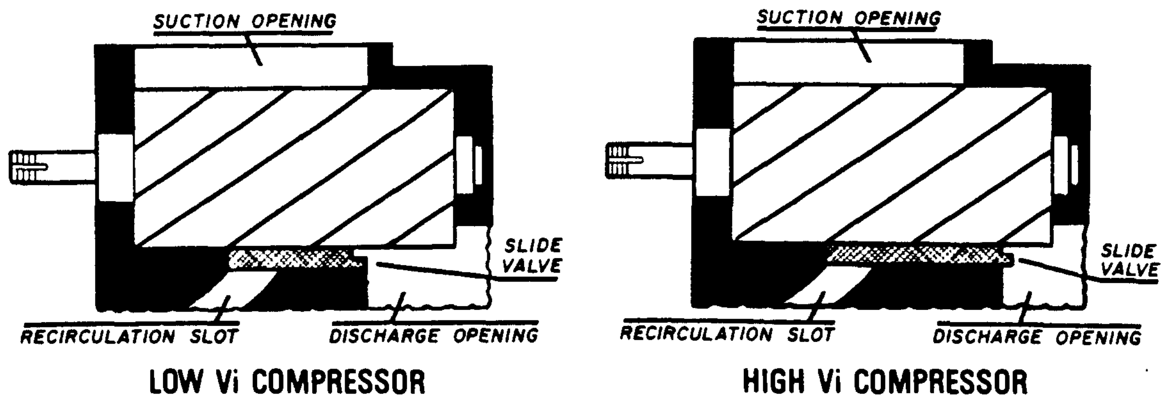


Figure 5

VIEW OF SLIDE VALVE FROM ABOVE

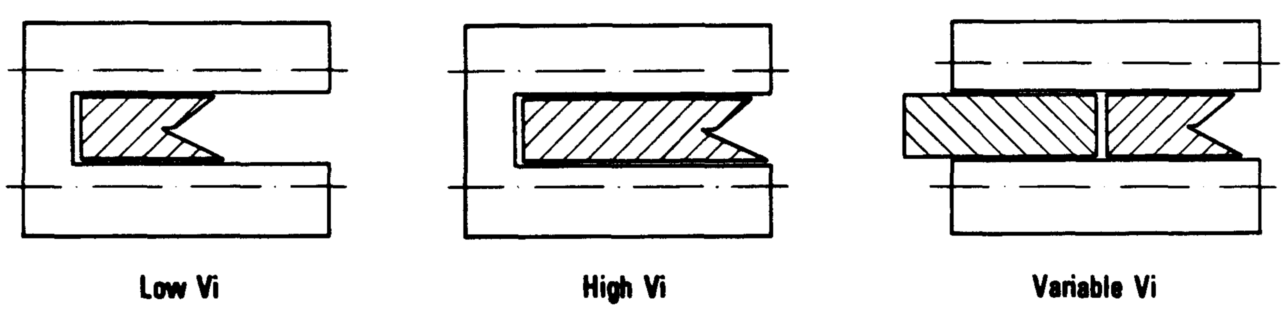


Figure 6

of the suction charge back to the suction cavity before compression begins. By moving the valve closer to the discharge the effective rotor length is reduced and capacity can be easily modulated without high power penalties. Slide valves of different lengths are employed to locate the radial discharge port and establish the desired Vi.

Figure 5 shows schematically two different length slide valves in a compressor. The short slide valve gives a low compressor Vi and the long slide valve gives a higher Vi. The moveable slide stop is shown schematically in Figure 6. Note that the slide stop and slide valve are independently moveable sections. By moving them both toward the discharge end of the machine with no recirculation passage, the compressor continues to run full load while being adjusted to higher and higher Vi, up to the limit imposed by the fixed axial outlet port.

A hydraulic control arrangement can be used to separately control the slide valve and slide stop. This allows the compressor to be adjusted while running, to the optimum Vi required by the system conditions. Thus, the compressor volumetric and adiabatic efficiency is increased over a wide operating range, as shown in Figure 7.

The greatest energy savings potential with variable Vi can be seen in a swing machine operation. For example, suppose a compressor is used as the high stage on a 2 stage system during the week at 95° F condensing and 20° F suction, and then is used single stage on the weekend at 95° F condensing and -20° F suction to hold a coldroom to some nominal temperature.

SCREW COMPRESSORS

EFFICIENCY IMPROVEMENT with VARIABLE VOLUME RATIO

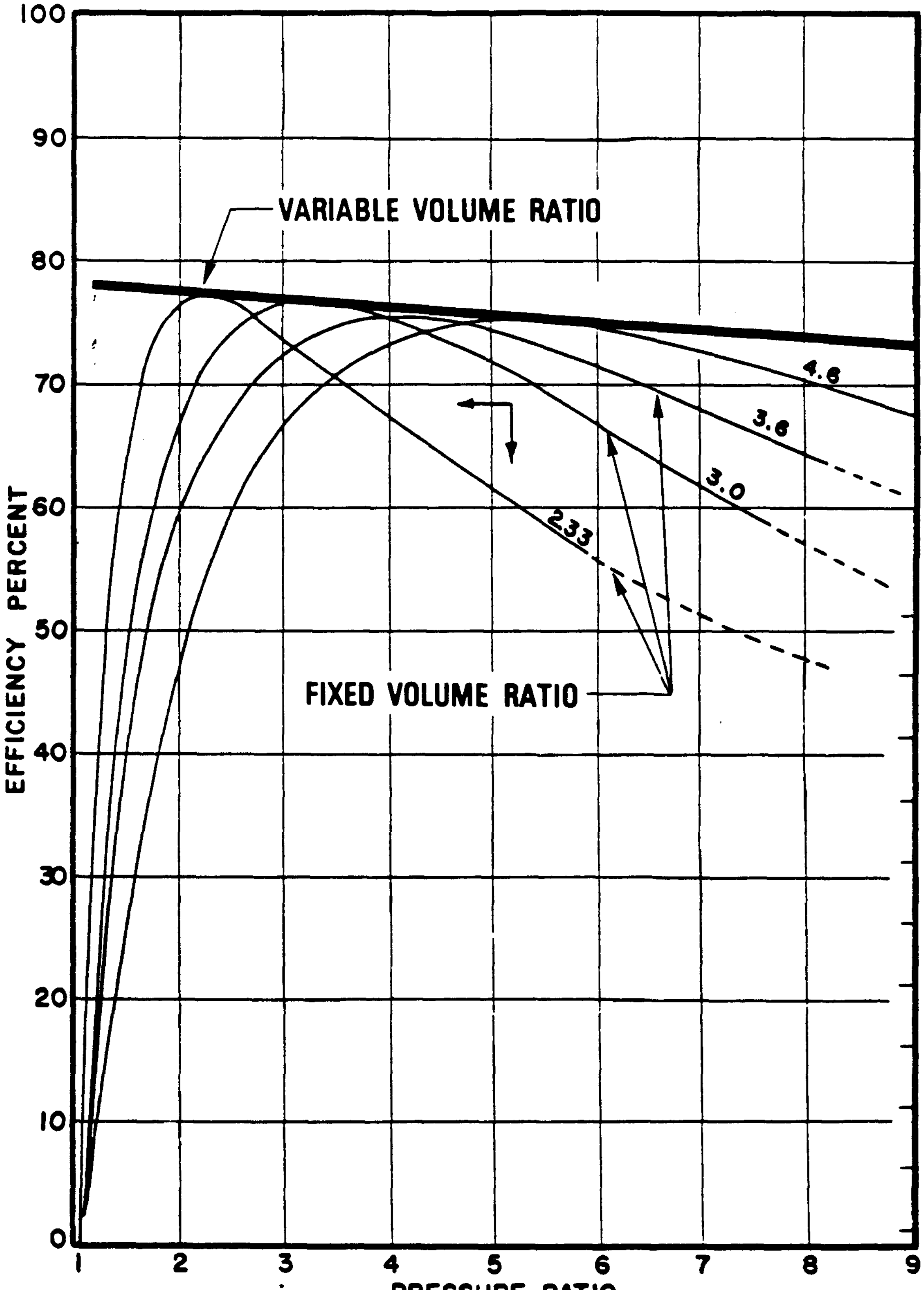


Figure 7

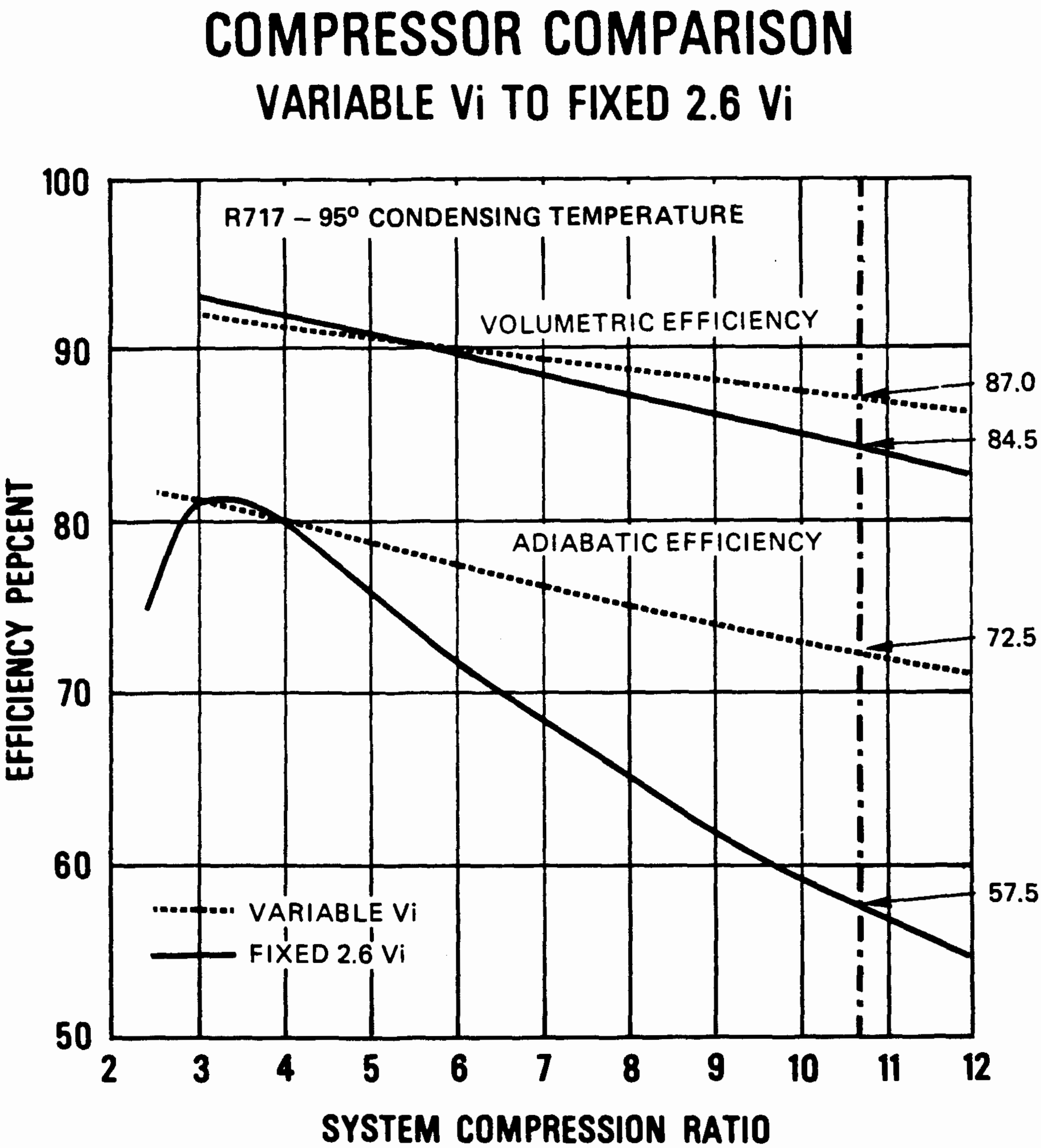

COMPRESSOR COMPARISON
VARIABLE Vi TO FIXED 2.6 Vi
R717 – 95° CONDENSING TEMPERATURE
VOLUMETRIC EFFICIENCY
ADIABATIC EFFICIENCY
87.0
84.5
72.5
57.5
VARIABLE Vi
FIXED 2.6 Vi
EFFICIENCY PEPCENT
100
90
80
70
60
50
2
3
4
5
6
7
8
9
10
11
12
SYSTEM COMPRESSION RATIO

Figure 8

Actual test data for a compressor with variable volume ratio is shown in Figure 8 plotted against a 2.6 Vi fixed volume ratio compressor of approximately the same swept volume. Note that at 95° F condensing 20° F suction, the compression ratio is 4:1 and performance is nearly identical. However, when this system operates part time at 95° F condensing and -20° F suction temperature, the compression ratio would be 10.7:1. At this condition the variable Vi compressor offers a 3% improvement in volumetric efficiency and a 26% improvement in adiabatic efficiency. This converts to 3% more usable refrigeration and 26% less horsepower.

The amount of improvement that can be obtained in any system will depend primarily on the expected fluctuation in system conditions. The greater the fluctuation, the greater the energy savings.

PART LOAD OPERATION

The manner in which full load Vi can be varied with the moveable slide stop is rather obvious from inspection of the design. However, part load operation becomes somewhat more complex.

The moveable slide stop and slide valve can be moved apart to provide a passageway between them to by-pass a portion of the suction gas back to the suction cavity. The width and location of this passageway can be controlled to reduce capacity and vary the Vi simultaneously within certain limits.

In a conventional slide valve equipped compressor, the internal suction and discharge

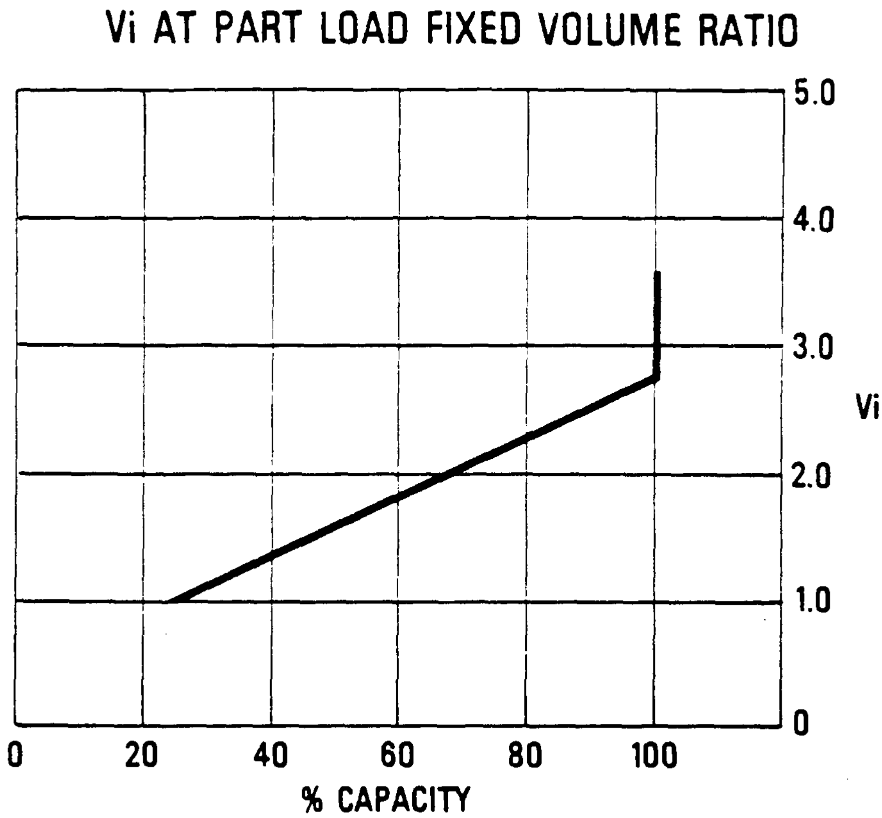


Figure 9

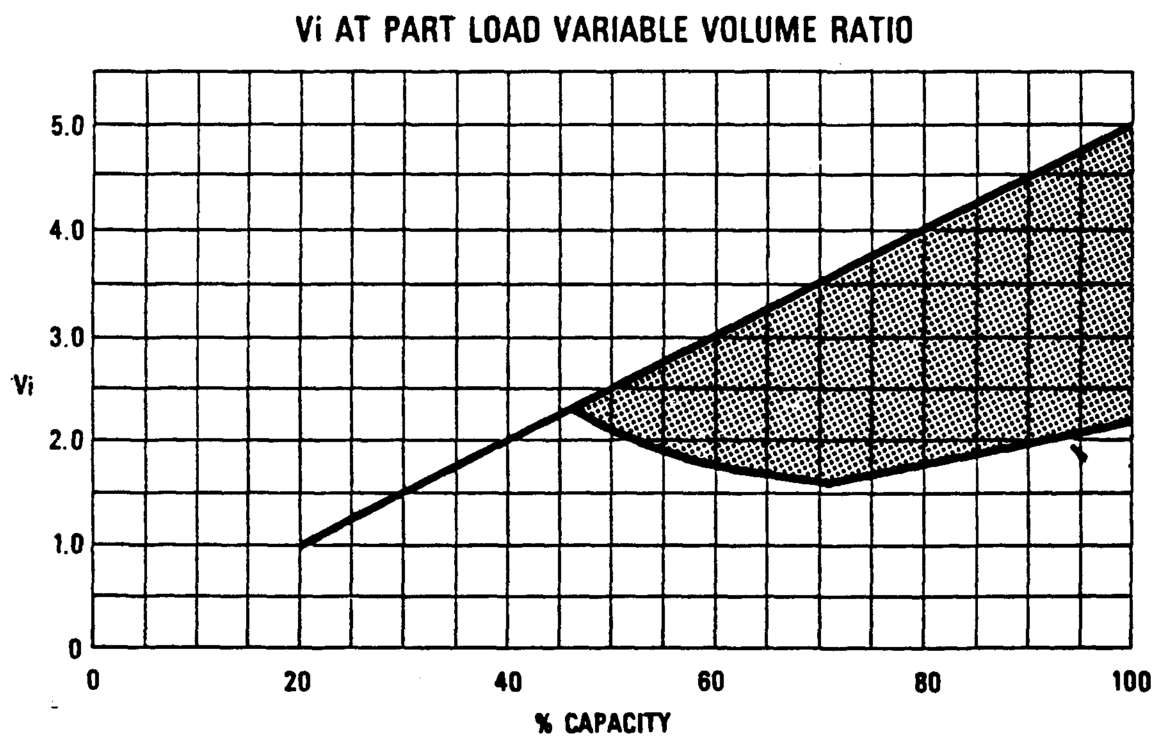


Figure 10

volumes are both reduced as the compressor unloads. Thus the ratio of the volumes, (the Vi) remains somewhat constant at the design Vi as the compressor unloads. See Figure 9.

The range of part load Vi adjustability with the moveable slide stop is shown in Figure 10. As the slide valve is unloaded on the variable Vi compressor with the slide stop fixed, the suction and discharge volumes are also reduced proportionately, roughly maintaining whatever full load Vi the compressor was running at that time.

If a higher Vi is desired, both slides can be moved further toward the discharge and the gap between them made smaller. Instead of allowing the recirculating gas to by-pass freely to suction, the gap is used as a throttle. The percentage of gas being recirculated is throttled from some slightly raised pressure back to suction pressure while the remainder of the gas charge still trapped in the threads is compressed from this slightly raised pressure to discharge pressure. In this manner the effective Vi can be increased. The losses associated with throttling a portion of the suction gas are insignificant by comparison to the savings obtained by operating at the correct discharge pressure without over or under compression.

As at full load, the efficiency improvement at part load over a fixed Vi compressor will depend upon how much the system pressures fluctuate, or how far they are from the design ratio. Figure 11 shows the percentage improvement possible at various system compression ratios compared to a fixed 2.6 Vi compressor.

PART LOAD EFFICIENCY IMPROVEMENT WITH VARIABLE Vi COMPARISON WITH FIXED 2.6 Vi COMPRESSOR

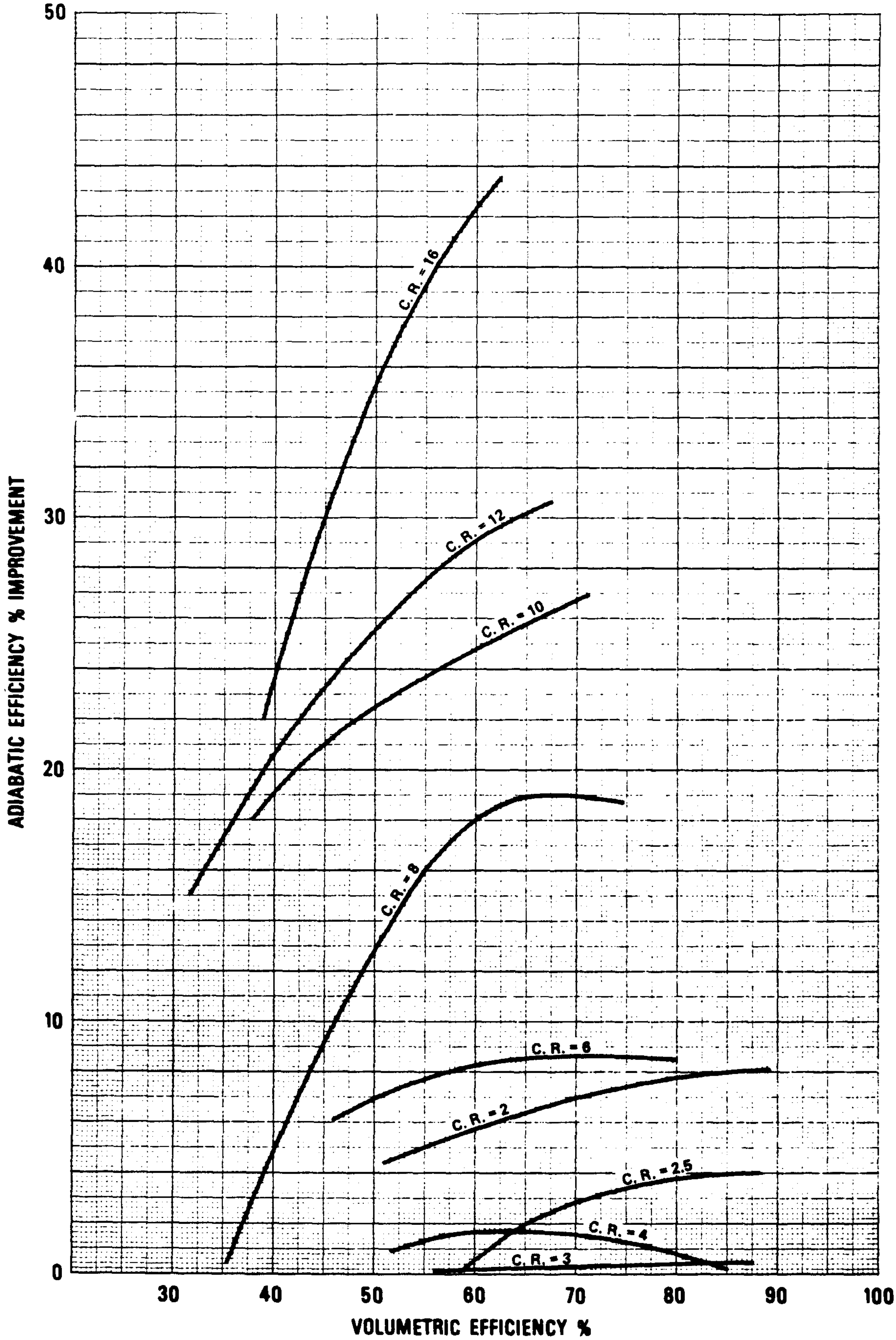


Figure 11

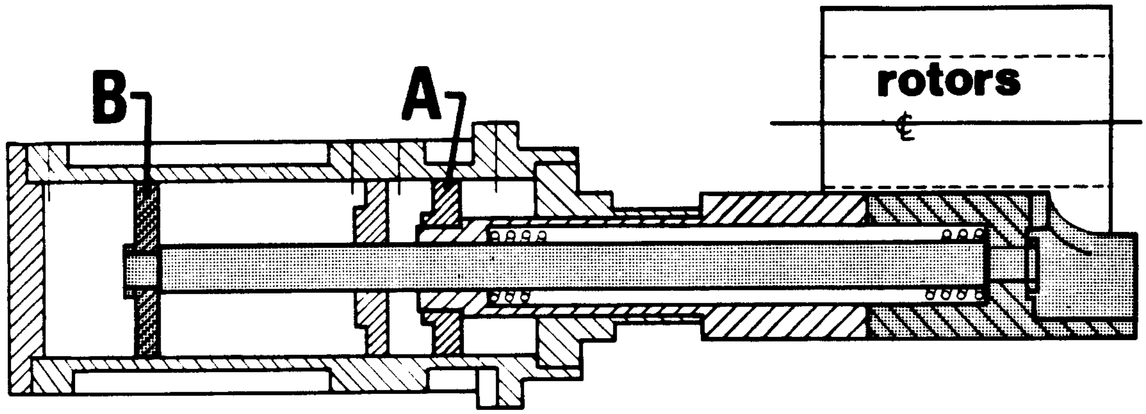


Figure 12

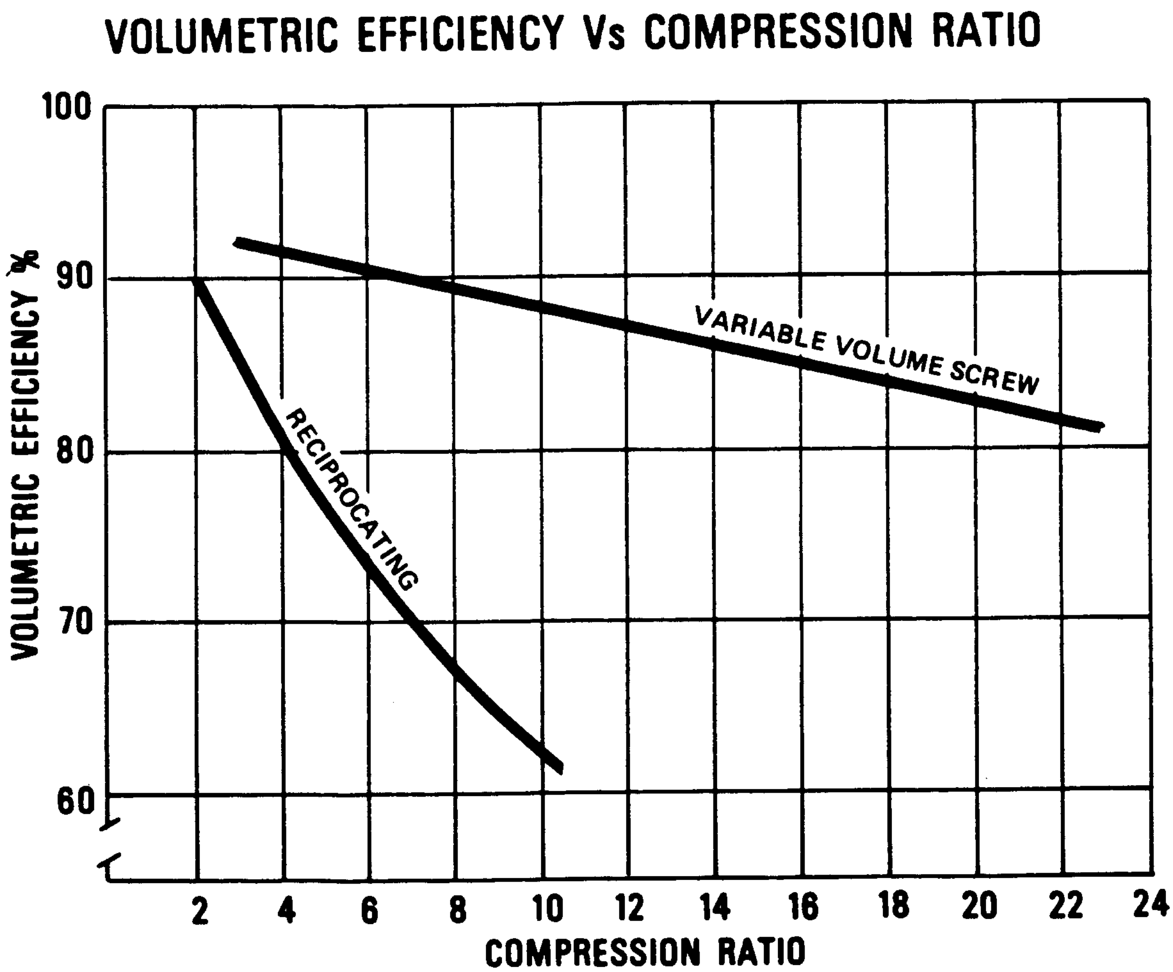


Figure 13

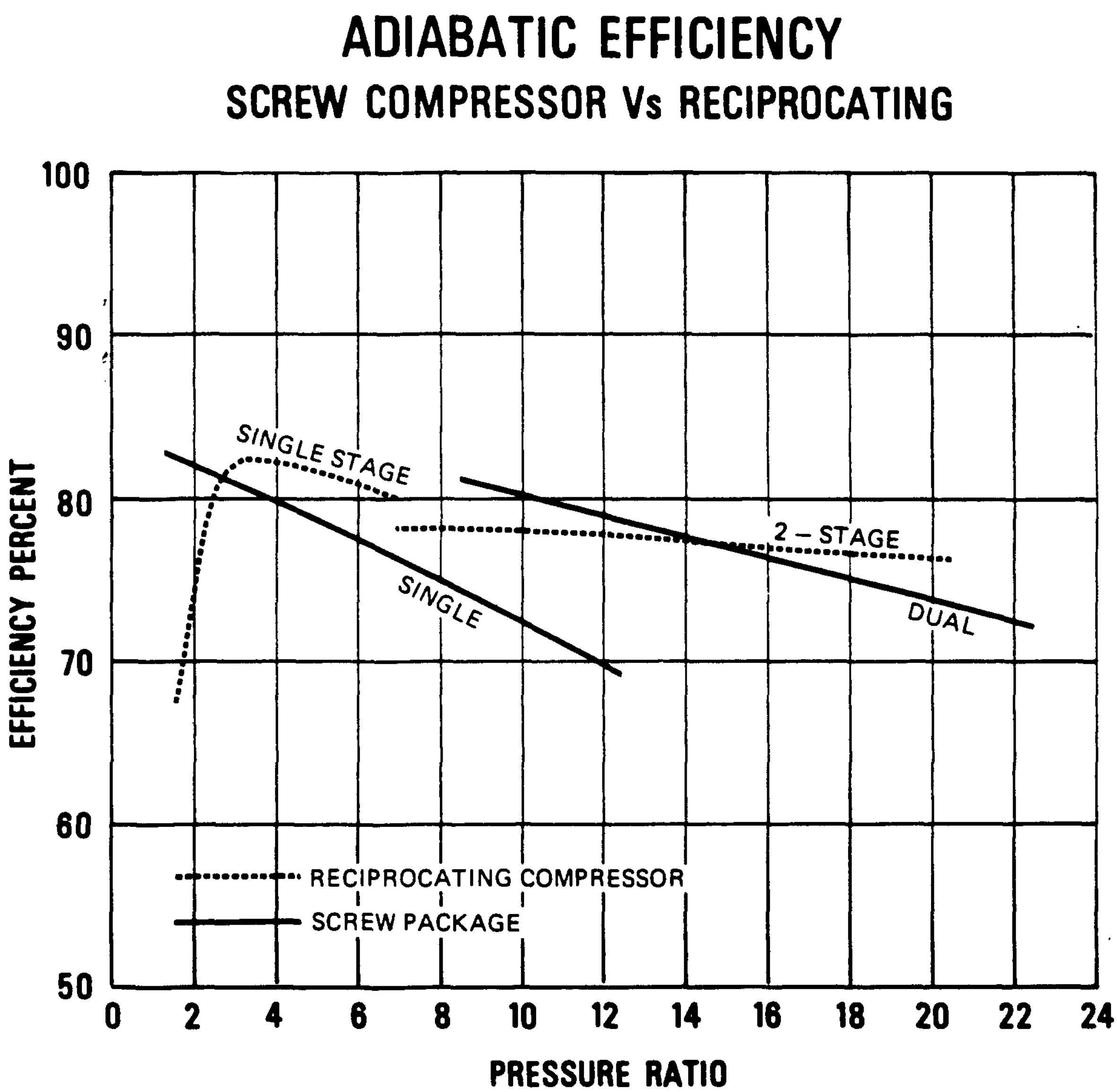

ADIABATIC EFFICIENCY
SCREW COMPRESSOR Vs RECIPROCATING
EFFICIENCY PERCENT
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
PRESSURE RATIO
SINGLE STAGE
SINGLE
2 – STAGE
DUAL
RECIPROCATING COMPRESSOR
SCREW PACKAGE

Figure 14

CONTROL ARRANGEMENTS

One arrangement for controlling the sliding members is shown in Figure 12. The slide stop is controlled by piston A in the above figure and the slide valve by piston B. High pressure oil admitted to the oil ports can be used to move and lock the sliding members in the desired positions. Solenoid valves are used to direct high pressure oil to the pistons or vent the cylinders to low pressure.

In order to operate the compressor at peak efficiency under full load and part load conditions, and with different refrigerants, complex control sequences must be used. It is believed that in order to obtain the efficiencies inherent in this design, microprocessor controls are almost mandatory. The microprocessor allows the user to obtain improved efficiency as the system pressures vary, whether a change is brief or sustained. This type of flexibility would be virtually impossible with conventional controls.

COMPARISON TO RECIPROCATING COMPRESSORS

During the development and testing of the variable volume ratio screw compressor it has become evident that this single development has greatly improved the competitive position of the double helical screw compressor against other types of compressors, notably reciprocating compressors. The installation and maintenance costs as well as package size of screw compressors have allowed them to replace reciprocating compressors in many installations. However, the adiabatic efficiency of reciprocating compressors has remained slightly above screw compressors on most refrigeration applications for many years. With the introduction of variable Vi this gap has not been completely eliminated but it has certainly been narrowed (see Figure 13 and 14). It is not expected that screw compressors will ever completely replace reciprocating compressors in the refrigeration field. However, as screw compressor performance improves, they will provide advantages on an increasing number of applications.

CONCLUSIONS

The development of the variable Vi screw compressor is a significant advance in compressor technology. The SRM double helical screw design has proven its durability and performance in the refrigeration market for roughly 25 years. With the added efficiency and flexibility of variable Vi the screw compressor has overcome one factor that has restricted its range of application.

This affords today's refrigeration system designers greater freedom in applying screw machines and provides lower energy costs for today's refrigeration users.