United States Patent [19]

Moody, Jr. et al.

[11] **3,885,402**

[45] **May 27, 1975**

[54] **OPTIMIZED POINT OF INJECTION OF LIQUID REFRIGERANT IN A HELICAL SCREW ROTARY COMPRESSOR FOR REFRIGERATION USE**

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[73] Assignee: **Dunham-Bush, Inc.**, West Hartford, Conn.

[22] Filed: **Jan. 14, 1974**

[21] Appl. No.: **433,418**

[52] **U.S. Cl.** **62/505;** 418/97; 418/201

[51] **Int. Cl.**..... **F25b 31/00;** F01c 21/06; F01c 1/16

[58] **Field of Search** 418/97, 201; 62/197, 228, 62/505

[56] **References Cited**

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Primary Examiner—John J. Vrablik
Attorney, Agent, or Firm—Sughrue, Rothwell, Mion, Zinn and Macpeak

[57] **ABSTRACT**

A helical screw rotary compressor within a refrigeration system is provided with a liquid refrigerant injection port opening into the working space of the compressor for injecting liquid refrigerant bled from the refrigeration system condenser at essentially compressor discharge pressure. The port is located at the optimum point of injection by determining; the minimum built-in volume ratio of the compressor, the rotor bore within which injection is to take place, the minimum and maximum system or operating compression ratios and by correlating these parameters with the pressure versus wrap angle plot for the selected compressor rotor bore. The acceptable injection point for the injection port is thus determined in terms of the wrap angle from the suction side of the compressor for the screw rotor of the selected bore where continuous pressure injection of liquid refrigerant occurs within the working space regardless of undercompression or overcompression of the working fluid without significantly compromising compressor efficiency.

4 Claims, 10 Drawing Figures

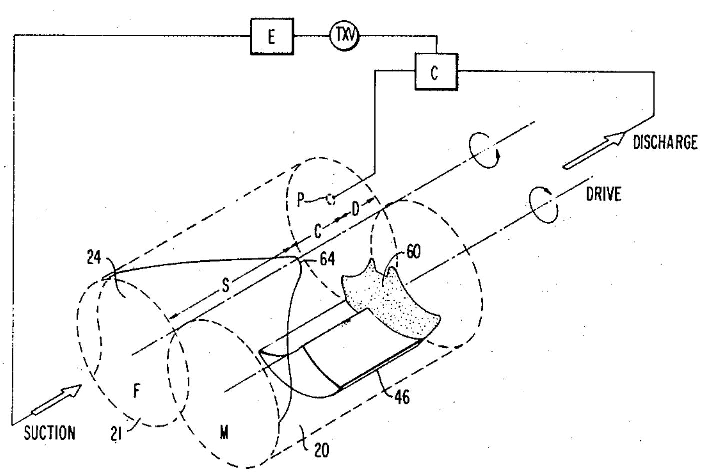
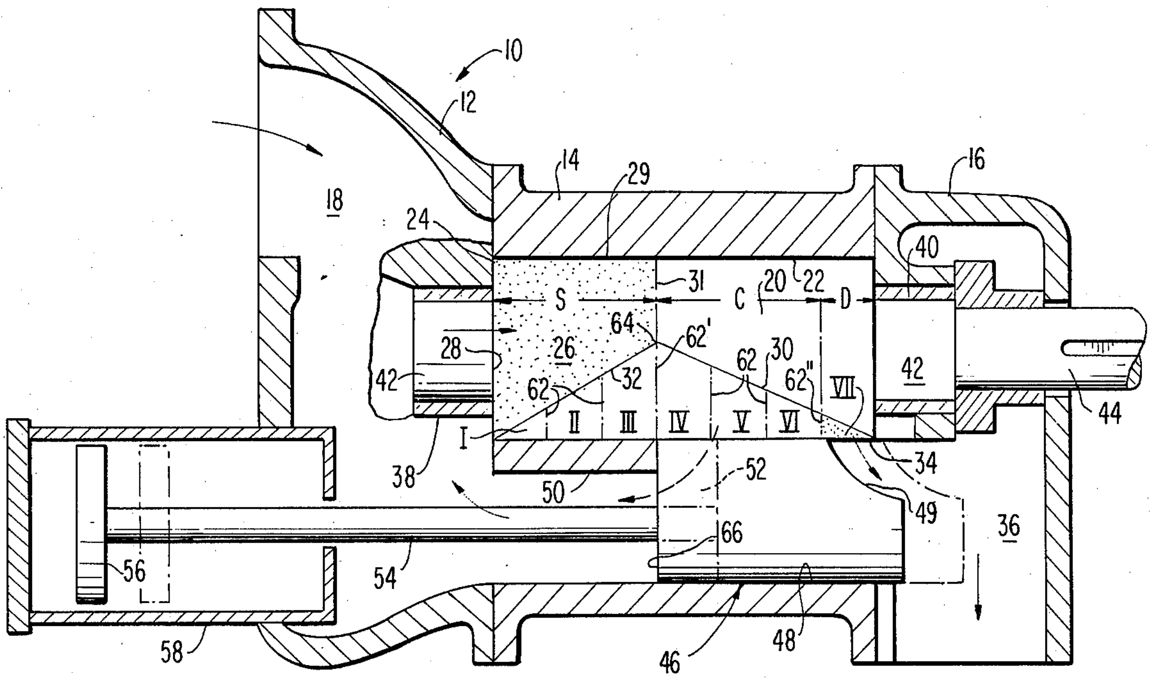


FIG.1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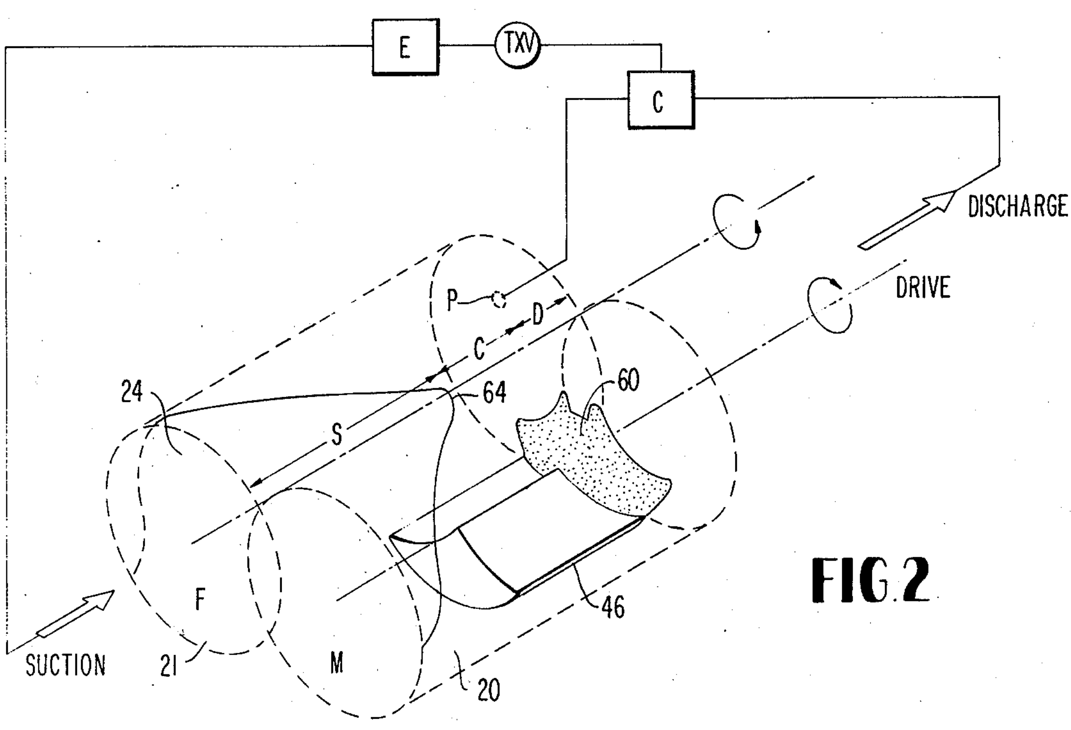
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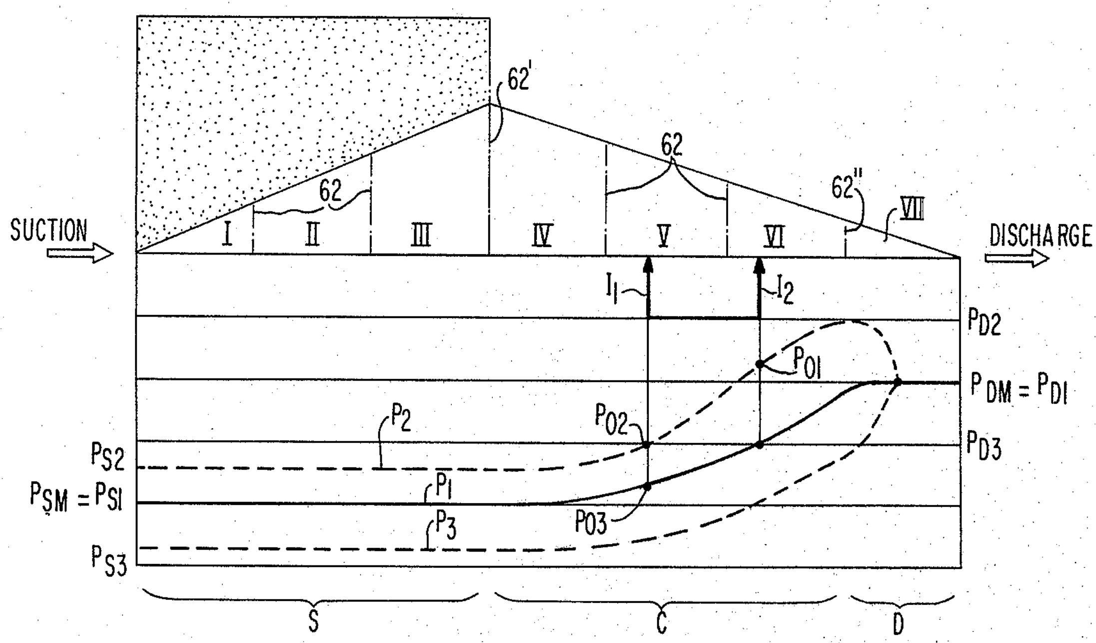
FIG.2

FIG. 3



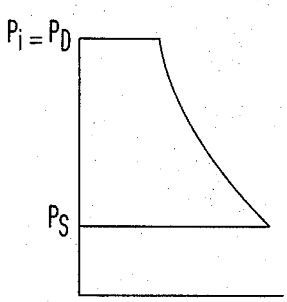


FIG. 4a

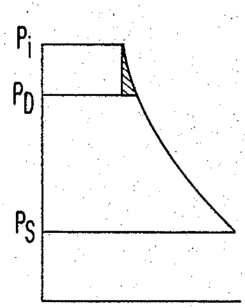


FIG. 4b

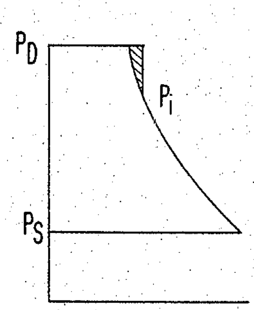


FIG. 4c

FIG.5

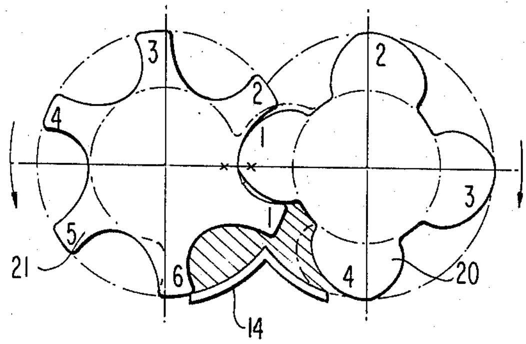


FIG.6

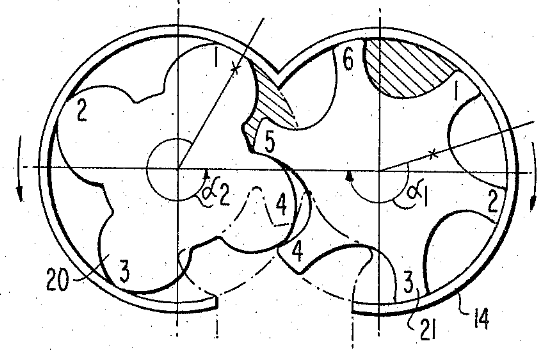


FIG.7

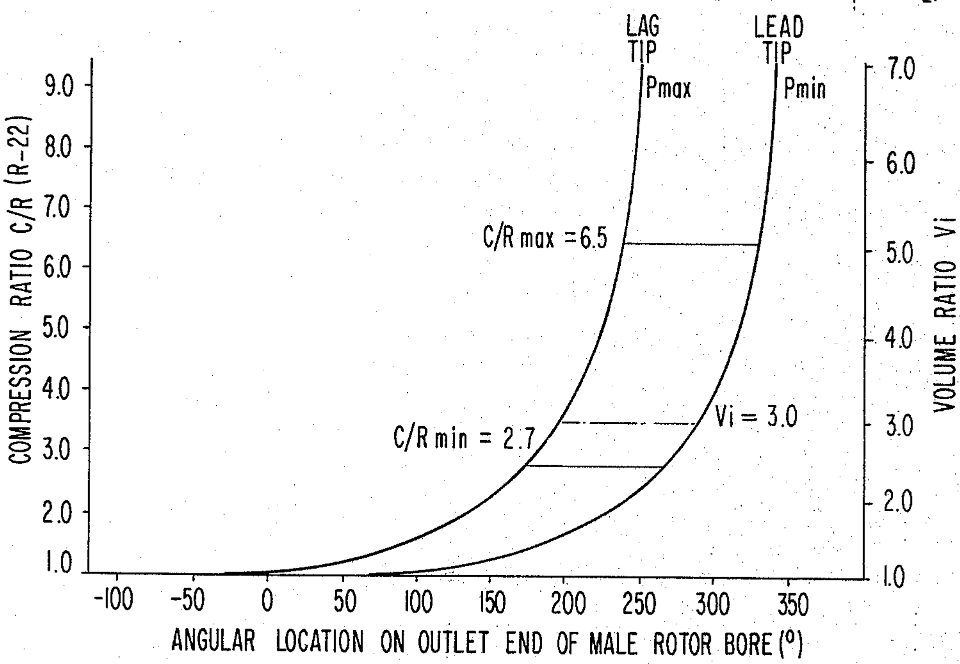
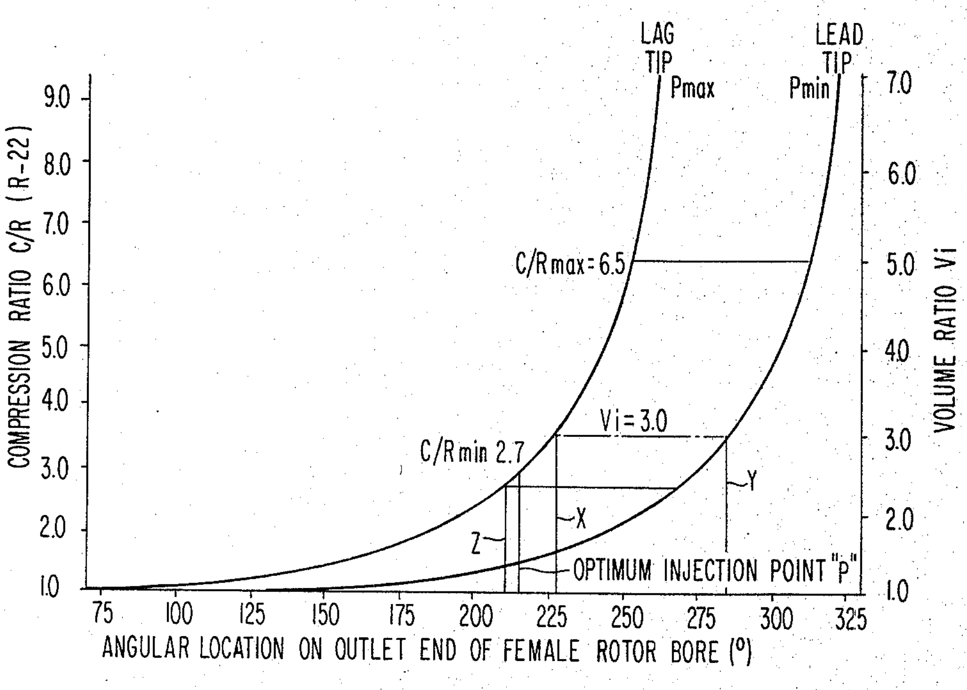


FIG.8



OPTIMIZED POINT OF INJECTION OF LIQUID REFRIGERANT IN A HELICAL SCREW ROTARY COMPRESSOR FOR REFRIGERATION USE

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to helical screw rotary compressors for operation within a refrigeration system, and more particularly, to a helical screw rotary compressor in which liquid refrigerant is injected within the compressor working space to cool the gaseous refrigerant working fluid during compression thereof.

2. Description of the Prior Art

The present invention relates to helical screw positive displacement rotary compressors of the oil injected or dry type, and more particularly, to such a compressor in which the point of injection of liquid refrigerant is optimized. Helical screw rotary compressors for compressing an elastic working medium generally comprise a housing or casing containing a working space formed by two parallel and intersecting bores with a low pressure or suction port at one end and a high pressure or discharge port at the other. Two intermeshing helical screw rotors are rotatably mounted in respective bores and, if desired, a slidable unload valve member may form part of the wall of the working space, whose position axially determines the capacity of the machine.

In an effort to eliminate the need for timing gears connecting the rotors for synchronized rotation, a liquid such as oil has been injected into the compressor for use as a seal closing the clearance space between the rotors and between the rotors and the housing to reduce blow back, as well as for directly cooling the working fluid being compressed.

Where the rotary screw compressor forms a part of a refrigeration or air conditioning system and the working fluid constitutes a typical refrigerant, it has further been proposed to inject liquid refrigerant into the compressor working chamber for cooling purposes, where high pressure liquid refrigerant is bled from the condenser in the refrigeration circuit and which is essentially at compressor discharge pressure. Controlled injection of liquid refrigerant in such a manner forms the subject matter of U.S. patent application Ser. No. 285,695 to Harold W. Moody et al., filed Sept. 1, 1972, entitled "Injection Cooling of Screw Compressors" and assigned to the common assignee, now U.S. Pat. No. 3,795,117.

Refrigerant liquid injection cooling of compressor eliminates the need for the usual oil cooler. Oil cooler heat injection load is usually a sizable percentage of the total heat rejection due to the amount of oil circulated. This eliminates the need for an oil cooler, piping and oil cooler control means.

The determination of that point within the compression cycle for porting the liquid refrigerant to the working chamber of the compressor is extremely difficult due to the variation in loading on the compressor, particularly where the screw compressor is not operating under a constant system or operating compression ratio. Where the screw compressor forms a component of the refrigeration or air conditioning system and where the refrigeration system loads vary, the system or operating compression ratio of the screw compressor varies with changes in suction pressure, and thus the compressor discharge pressure varies.

SUMMARY OF THE INVENTION

It is an object, therefore, of the present invention to provide a helical screw rotary compressor in which the point of liquid refrigerant injection to the compressor working chamber is optimized to the varying system operating parameters.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional, elevational and partially schematic view of a helical screw rotary screw compressor incorporating the liquid refrigerant injection scheme of the present invention.

FIG. 2 is a schematic, perspective view of the intermeshed male and female helical screw rotors of the compressor of FIG. 1.

FIG. 3 is a plot of the pressure curve for the screw compressor of FIG. 1 and a two dimensional representation of the variation of the change of the volume of the compressed working fluid within the compressor from suction to discharge.

FIG. 4*a* is a PV plot for the screw compressor of FIG. 1 under ideal conditions.

FIG. 4*b* is a PV plot for the compressor of FIG. 1 under overcompression conditions.

FIG. 4*c* is a PV plot of the compressor of FIG. 1 illustrating undercompression conditions.

FIG. 5 is an inlet end plane representation of the intermeshed screw rotors showing the creation of a closed pocket with the suction port superimposed thereon.

FIG. 6 is an outlet end plane representation of the compressor of FIG. 1 illustrating the angular relationship between the intermeshed screw rotors and one of the chambers sealed off from the suction and discharge sides of the compressor with the axial discharge port superimposed thereon.

FIG. 7 is a pressure versus male rotor wrap angle plot in the outlet end plane for the compressor of FIG. 1.

FIG. 8 is a pressure versus female rotor wrap angle plot in the outlet end plane for the compressor of FIG. 1.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Necessary to a complete understanding of the parameters involved in a determination of the optimum point for the injection of a liquid refrigerant within a helical screw rotary compressor, is a visual understanding of the manner of change in volume of the gas chambers which are formed between the helical screws and the housing. The helical screw rotary compressor constitutes a positive displacement compressor in the same fashion of the typical reciprocating type compressor wherein a piston reciprocates within a cylinder toward and away from the head, with the gas pocket in this case being of maximum size at the end of the suction stroke, wherein the piston is at bottom dead center and with the chamber being compresssed to its smallest volume as the piston reaches top dead center under the compression stroke. Such volumetric change may be visually appreciated quite easily. Further, in the typical rotary compressor of the sliding vane type, the rotor is eccentrically mounted with respect to the cylindrical chamber within which it rotates and the sliding vanes move radially in the rotor with a chamber determined between the outer cylindrical casing and the cylindrical rotor along with any two vanes and being quite large on the suction side of the machine, but being reduced considerably in volume after approximately 180° rotation, wherein that chamber opens up to the discharge port. Again, the pictorial presentation of the change in volume of the chamber, that is, the reduction in size from suction to discharge may be readily seen in two dimensions by a simple sectional view at right angles to the axis of rotation of the rotor in the case of the sliding vane type rotary compressor and a sectional view parallel to the reciprocating axis of a typical reciprocating type compressor.

In contradiction, the screw compressor, while having fewer moving parts and constituting a rather simple operational concept, is not capable of providing a simple two dimensional presentation in terms of the change in volume of the individual gas chambers determined by the helical screws and the housing, but requires considerations of depth or third dimension.

Reference to FIG. 1 illustrates in two dimensions and in partial schematic form, a sectional view of a typical helical screw compressor. In pictorial form, the compression of the working fluid in gaseous form, moves from the low pressure suction or intake side of the screw compressor to the high pressure or discharge side of the same. In this respect, the typical screw compressor indicated generally at **10**, to which the present invention has application, comprises a three part housing or casing as at **12, 14** and **16**, the housing **12** being the suction side of the machine and being provided with an intake passage **18**, housing **12** leading to the center housing **14** which houses the two intermeshed helical screw rotors, only one of which is shown in block form at **20** in the sectional elevational view of FIG. **1**. Housing **14** therefore contains the working space in the form of two intersecting bores, with bore **22** housing rotor **20** and wherein suction passage **18** terminates in a low pressure suction port at end plane **24**. The suction volume is illustrated schematically in quadrilateral form at **26** by lines **28, 29, 31** and **32**. A high pressure combined axial and radial discharge port at **34** at the opposite ends of the intermeshed screw rotors opens up into the compressor discharge passage **36** forming the principal portion of the housing **16**. Each intermeshed screw rotor, such as helical screw rotor **20** is supported for rotation by bearings **38** and **40**, receiving the projecting ends of shafts **42** within respective housings **12** and **16**. The screw rotor **20** in this case is positively driven by drive shaft **44** which may form an integral part thereof. The drive shaft **44** is driven by a prime mover (not shown). Pertinent to the present invention, although conventional to rotary screw compressors, is an unloader valve indicated generally at **46** which is mounted for reciprocation within a bore **48** of central housing **14**, the valve **46** defining part of the working space further defined by the bores such as bore **22** within casing **14** and the screw rotors themselves such as rotor **20**. In this case, the unloader valve **46** is provided at its discharge end with a scalloped or recessed leading edge **49** forming the radial discharge section of the discharge port **34** of the compressor. The unloader valve **46** is of such axial length and cooperates with a stationary wall portion **50** of housing **14**, which acts as a fixed stop for the unloader valve **46**, to form a bypass or unload port **52** which leads directly to the suction or inlet passage **18**. An unloader shaft **54** couples the unloader valve **46** to a piston **56** mounted for reciprocation within unloader cylinder **58** which is fixed to the housing section **12**, the piston **56** being movable between the full line position shown and the dotted line position by the controlled application (not shown) of pressurized fluid such as lubricating oil bled from the lubricating system of the compressor to shift the unloader valve between load and unload positions as determined by refrigeration system load demand.

FIG. **1** illustrates in two dimensions the schematic change in volume of the gas chambers as formed between the helix of the rotors and the housing, this may be perhaps best appreciated by further reference to FIG. **2**, which shows the essential change in volume of the gas volume open to suction at the left hand end and top of the intermeshed screws and the discharge volume at the lower right hand end of the compressor intermeshed screw rotors. In this case, both the male helical screw rotor **20** and the female helical screw rotor **21** are shown as intersecting cylinders in dotted lines, with the intersecting cylinders indicative of the intermeshed lands and grooves of respective rotors. In the schematic, perspective representation of FIG. **2**, the unload valve **46** is shown as corresponding to full load position of FIG. **1**. Immediately to the right of the unload valve **46**, the volume of the compressed gas open both to axial and radial porting leading to the discharge passage is indicated at **60** and takes an irregular form due to the configuration at the discharge end of the intermeshed screws. The refrigerant in conventional fashion and in the manner of U.S. Pat. No. 3,795,117 travels in a closed loop from compressor discharge back to compressor suction. The loop, FIG. 2, includes in order; condenser C where it is condensed (giving up heat), thermal expansion valve TXV where the liquid refrigerant expands, and evaporator E where it takes up heat by heat of vaporization.

In the two dimensional illustration of FIG. **1**, the intermeshed screw rotors form individual gas chambers or pockets separated by vertical seal lines such as **62**. The chambers are identified in Roman numeral fashion at I, II, III, IV, V, VI and VII; chambers I, II, and III being open to suction and the vertical seal line **62'** being the suction porting seal off line, that is, this line intersects lines **30** and **32** at point **64** corresponding to the same point **64** in FIG. **2** where none of the gas chambers to the right are open to suction. Compression occurs in a downstream direction, that is, to the right of cutoff point **64** and seal off line **62'**. In the schematic illustration, compression occurs within gas chambers IV, V and VI, with chamber VII being open to discharge. In this respect, the sequence of suction, compression and discharge is indicated longitudinally by headed arrows designated S, C and D respectively in both FIGS. **1** and **2**. In terms of the unloading valve **46**, absent that valve, the sequence remains as shown. However, the shifting of the unloader valve **46** from the fully loaded, solid line position where it abuts section **50** of the housing, FIG. **1**, causes, as the valve moves away from housing **50**, some of the gas which would normally be compressed to be returned to the suction side of the machine by way of bypass port **52** and bore **48** back to the suction passage **18**. This reduces the amount of the gas being compressed, schematically as identified in FIG. **1** by the position of the left end **66** of the unloader valve **46** relative to diagonal line **30** which shows the change in chamber or pocket volume for chambers IV, V, VI and VII during compression. As seen further by the further figures within this application, the size or volume of the gas chambers IV–VII are reduced thus causing compression of the gas contained therein as the chambers are moved in sequence from the suction zone to the discharge zone and in that respect obviously a new gas chamber replaces chamber I as it moves to position II as result of rotation of the screws in the manner illustrated in FIG. **2,** due to the direct meshing of the helical lobes of respective rotors.

From FIGS. **1** and **2,** it is obvious that when the male and female rotors are placed within a housing having fixed suction and discharge ports **24** and **60,** they form gas passages and gas chambers between the helix of the rotors and the housing, thus accommodating the suction gas which is then compressed due to volume reduction of the gas chambers as the engaged, mated rotors turn in a synchronized manner to produce gas pumping and compression while moving the refrigerant gas from the suction side to the discharge side at opposite ends of the housing. The complexity in clearly defining the seal lines and the porting within the compressor housing hinders attempts to define the narrow range of optimum location of the liquid refrigerant injection port, which is the aim of the present invention.

A gas chamber created by the intermeshed male and female rotors and the housing **14** may be more fully appreciated by reference to FIGS. **5** and **6.** The male rotor rotates at a speed one and one-half times that of the female rotor. A horizontal center line passes through the axes of both rotors, FIG. **5,** and intersects the contact point between lobe **1** of the male rotor **20** and a point between lobes **1** and **2** of the six lobed female rotor **21** as viewed from the suction or inlet end. The gas chambers formed thereby wrap 200° about female rotor **21** toward the discharge side of the machine, as indicated by the angle α_1 while correspondingly, that chamber portion of male rotor **20** wraps to angle α_2 of 300° about the rotor axis as indicated on the outlet end plane representation of FIG. **6.**

The screw compressor such as that illustrated in the figures has a fixed built-in volume ratio $V_i = V_S/V_D$ where V_S equals the volume of the suction and V_D equals the volume at discharge. The screw compressor pressure ratio P_i, in this case, is equal to V_i^k where k is the ratio of specific heat for the fluid being compressed. This relationship is derived from the basic thermodynamic gas laws where:

$$P_1 V_1^n = P_2 V_2^n$$

$$\frac{P_2}{P_1} V_2^n = V_1^n$$

or

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n$$

Where ***K*** or n is ratio of specific heat.

For the compressor

$$VL = \frac{V_1}{V_2}$$

$$\text{And } P_i = V_i^n = \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n$$

Therefore, if a positive displacement compressor has an inlet volume V_1 and for a given gas, has a known k or n (ratio of specific heat), the corresponding pressure or charges of pressure can be calculated for changes of gas volume. In this case, the working fluid of the compressor may comprise refrigerants such as **R22,** where k equals 1.15, or ammonia (NH_3) where k equals 1.29.

As may be more clearly seen by reference to FIGS. **4***a*, **4***b* and **4***c*, where P_i equals P_D/P_S, the system compression ratio C/R matches the compressor pressure ratio under ideal conditions, that is, $P_i = P_D/P_S = C/R$, as evidenced in FIG. **4***a*. A system compression ratio C/R is determined by relationships imposed by the system evaporator and condenser. This involves the heat exchange capability of each component. These relationships can change as follows:

Heat input evaporator increases	C/R decreases
Heat input evaporator decreases	C/R increases
Heat input condenser increases	C/R increases
Heat input condenser decreases	C/R decreases

The positive displacement compressor with fixed gas porting operates with a fixed built in volume ratio and pressure ratio. Gas entering at pressure P_1 is discharged at P_i. The compressor compression ratio is fixed and is different from the system C/R that may be imposed by the system.

If the system components are sized exactly for the evaporator and condenser loads the system will operate at a certain C/R P_D. If the compressor V_i and resulting P_i are correct, the compressor and system can operate with the compressor C/R and the system C/R being equal.

System	$P_D/P_1 = C/R$
Compressor	$P_1 \times P_i = P_D$
	$\frac{P_D}{P_1} = P_i = C/R$

Turning to FIG. **4***b*, this is the situation where overcompression occurs and wherein the compressor pressure ratio P_i is greater than the system compression ratio C/R which is equal to P_D/P_S. Overcompression occurs and lost work due to overcompression is graphically seen by the shaded area within the PV diagram and wherein opening up of a given chamber within the compressor to discharge drops the pressure within that chamber to discharge pressure which is significantly lower.

Turning to FIG. **4***c*, the effects of undercompression are seen where P_i is less than $P_D/P_S = C/R$, and wherein the shaded area within that plot is indicative of the lost work due to undercompression with the discharge having to further compress the working fluid within the compressor at the point where the chamber opens up to discharge pressure P_D.

The effect of undercompression, overcompression with respect to that point of injection may be appreciated by reference to FIG. **3,** which constitutes a composite of a graph of pressure across the compressor from suction to discharge and a schematic representation of the change in volume from the suction side to the discharge side of the compressor.

FIG. 3 further illustrates overcompression and undercompression at the dash line P_2 and dash line P_3 respectively.

Overcompression occurs when the suction rises from P_{S1} to P_{S2}, then the discharge pressure reached internally due to the built-in P_i is P_{D2} within chamber VI as it opens up to the discharge port **34** because the built-in P_i is equal to P_{D2}/P_{S2}. Hence the Δ_P equals P_{D2} minus P_{D1}, causing gas expansion to the discharge porting and the drop as indicated by the drop in curve P_2 to manifold pressure P_{DM}.

Undercompression occurs when the suction drops from P_{S1} to P_{S3}, then due to the fixed P_i which is equal to P_{D3}/P_{S3}, the discharge pressure exposed to the discharge port **34** is P_{D3}, and since the manifold pressure is higher, the Δ_P is equal to P_{D1} minus P_{D3} and is compressed further by back-flow.

The effect of both overcompression and undercompression and the limitation provided by the unloading valve on the optimum liquid injection zone may be seen from further reference to FIG. **3.** In this figure, the liquid refrigerant injection zone or injection window is identified as the zone existing between the two vertical arrows I_1 and I_2 and are superimposed again on a two dimensional schematic representation of the compression of the working fluid in the nature of the upper part of composite FIG. **3,** while the pressure curves for the working fluid appear at the bottom of that figure. The liquid refrigerant injection zone, as partially defined by arrow I_1 lies just to the right of suction seal line **62,** for gas chamber IV associated with the lead tip, approximately one seal line away from the suction seal off **62'**.

While the injection zone constitutes a zone within the compression region C of the screw compressor, FIG. **3,** and while chamber IV just before lag tip clears a position where it closes off chamber IV to suction, liquid refrigerant injection could occur just before cutoff, since due to the high velocity rotation of the screw rotors, as liquid is injected into the chamber, the kinetics of the machine require that the chamber be immediately thereafter sealed from the suction side and the gas within the same starts to be compressed as the chamber moves toward the point wherein it opens up to the discharge side of the machine. Thus, in similar respect, the vertical arrow I_2 indicates the other possible limit in the angular position of the liquid refrigerant injection port for the machine, that is, injection would have to cease just before the chamber opens up to the discharge side. When the built-in ratio P_i matches the operating pressure ratio C/R, the theoretical maximum zone as defined by vertical arrows I_1 and I_2 may apply. However, the zone is narrowed when overcompression occurs, since the liquid refrigerant pressure as defined by the compressor discharge pressure, is about equal to P_{DM} which is the discharge manifold pressure (generally equal to refrigeration system condenser pressure from which the liquid refrigerant for injection is supplied) and must be higher than P_{O2}; the point P_{O2} on the pressure graph of FIG. **3** being that pressure within the chamber at the time of liquid injection as the result of overcompression due to the higher suction pressure P_{S2} than theoretical suction pressure P_{S1}. Assuming injection occurs at the high pressure end of the injection zone where the working fluid pressure as a result of overcompression has reached the value P_{O1}, no injection will occur if the assumption occurs that P_{O1} is higher than P_{DM}. The maintaining of continuous liquid refrigerant flow is imperative to proper cooling and operation of the machine. However, the liquid refrigerant injection is subject to pressure variants following the internal gas pressure at the point of injection and causes liquid flow variances. As shown in FIG. **3**, there may be a drop in suction pressure providing undercompression requiring back-flow compression to P_{DM}.

From the above description, many factors enter into determination of the optimum point of liquid refrigerant injection for various built-in and operating compression ratios for the screw compressor. Various steps must be followed. First, it must be established what the minimum built-in either in terms of axial compression ratio or radial compression ratio is for the particular compressor under consideration.

In FIG. **1** item **34** is the radial discharge port and in FIG. **2** item **60** is the axial discharge port.

If the radial discharge port and axial discharge port are set up so that gas starts to discharge from both at the same instant, the radial and axial compressor ratio are the same. If gas begins discharging from the radial port first, the radial compression ratio is lower than the axial compression ratio. Gas will begin to discharge from the axial port after the compressor rotors have turned slightly further.

The true compressor compression ratio is defined, as the compression ratio determined by the port that has the lowest theoretical "built-in," in other words, the port that begins discharging gas first.

In compressors without slide valves, the radial and axial compression ratio are usually the same. In compressors having slide valves, the radial port usually leads the axial (radial begins discharging before axial) in order to have an improved part load operating power characteristics. For example, if it has been determined that V_i radial is 3.0 for instance and V_i axial is 3.3, the minimum V_i is 3.0 and this is the figure to be used in further calculations. It is this parameter which must be later applied to the plot of either FIG. **7** or FIG. **8.**

Step number two is the establishment of which rotor bore liquid is to be injected into. This step determines the selection of the plot of FIG. **7** or FIG. **8** as the case may be.

Injection may be radial in a selected rotor bore or axial in an end plane.

The next step is the establishment of maximum and minimum system or operating compression ratios C/R as defined by the liquid refrigerant or other working liquid, and other parameters related to the pressure and temperature of the working fluid at both the intake or suction side of the machine and the discharge side. For instance, if refrigerant R22 is employed as the working fluid, and based on a minimum of 40° F. saturated suction and 105° F. saturated discharge, the minimum compression ratio is 2.7, while for the same refrigerant at 20° F. saturated suction and 145° F. saturated discharge, the compression ratio C/R maximum is 6.5. This is an example for a normal range air conditioning (40°/105°) and commercial duty (20°/145°) range of operation. Systems can be set up for almost any operating condition, however, the economics of operation KWH/Ton or EER will dictate tolerable system arrangements. These values are then superimposed on the plots of FIGS. **7** and **8.**

Reference may now be made to the pressure curves of FIGS. **7** and **8** for a given compressor and working fluid. Depending upon whether injection occurs within the male rotor bore or the female rotor bore, the next step is the drawing of horizontal lines on the selected pressure curve at positions representing:

C/R Max	=	maximum compression ratio
V_i	=	minimum built-in
C/R Min	=	minimum compression ratio

Based on these parameters, the next step is picking the angle at which injection is to take place. Injection must not take place at an angle that exposes the injection port to the minimum built-in pressure ratio of the compressor, that is, for example, assuming that female bore injection has been selected by step two and a minimum built-in of 3.0 as determined by step one, injection should never take place at an angle in excess of 286°. Vertical lines *X* and *Y* which emanate from the compressor built in line $V_i = 3.0$ at P_{max} and P_{min} plots where the lag tip and lead tip of the lobes defining a given closed compressor thread indicate two unacceptable areas for liquid refrigerant injection. Vertical line *Z* which intersects the lag tip plot line at the point where the horizontal line indicated minimum operating compression ratio for the compressor and system defines the minimum injection point *P* in terms of the angular location on the outlet of the selected female rotor bore. The rationale for this selection follows.

If the injection angle exceeds 286°, line *Y*, injection will never take place, since at 286° the pressure of the working fluid as result of compression will be such as to be higher than the pressure of the liquid refrigerant to be injected.

If on the other hand injection occurs between 226° line *X* and 286° line *Y*, injection will be intermittent, and reverse injection flow can occur if the compressor operates near its lowest designed compression ratio 2.7 as defined by step three, which can damage the injection modulation valve as well as limiting liquid injection flow. However, below 210°, line *Z* injection is assured, since the pressure of the liquid refrigerant within the line insofar as the liquid refrigerant is concerned, is well above that within the particular chamber in which injection is occurring.

With these considerations in mind, the low end of the overcompression range (210° to 226°) forms the injection window or zone for female rotor bore liquid refrigerant injection.

Picking an injection point in terms of an injection angle equal to 215° line *P* places it above the very minimum, that is, it would prevent injection into a chamber open to the suction side of the machine. Further, since injection is not an absolute necessity at low compression ratios, intermittent flow when operating at low compression ratios, would probably suffice. When injection occurs at an angle between 226° and 286°, injection is intermittent, that is, permitted during a portion of the rotation, and as rotation continues the pressure in the chamber becomes excessive to that of the bleed line and liquid refrigerant flow within the injection line reverses so that the net flow is minimal. Reverse flow may also occur in the injection line if the compressor operates near its lowest design compression ratio, that is, at 2.7. It must be noted that the closer to the chamber pressure that liquid injection takes place, the more expanded refrigerant will work its way back to the suction side and decrease the volumetric efficiency of the compressor. The closer to suction injection occurs, the greater the penalty to performance of the machine, so a compromise must lie within the injection window of from 210° to 226°. However, when the machine is operating at low compression ratio, such as the minimum 2.7, intermittent flow at low compression ratios is apparently sufficient to properly cool the machine. In the 210° to 286° region, the saturated pressure which is condenser or receiver pressure, is an acceptable pressure value above the injection port pressure. The measurable injection port pressure tends to be the minimum pressure theoretical at the injection port as determined by the compressor parameters plus one-third the difference between the minimum and maximum pressures from the graph of FIG. **8** and ΔP available is the discharge pressure minus the injection port pressure, assuming there are no pressure losses in the discharge line, the liquid injection line and the TX valve. In reference to FIG. **8**, the pressure of the trapped gas that is being rotated and compressed by the rotor is defined by the rotor turning angle and the theoretical pressure that exists at the lag rotor tip or lead rotor tip. The average pressure for a given rotor lobe is a pressure between lead tip and lag tip and about one-third distance above lead tip curve. ΔP is a term used for determining pressure difference available in order to have liquid flow into the compression chamber as described below, keeping in mind that unless the pressure of the liquid refrigerant to be injected is above that of the closed thread receiving the same, no liquid refrigerant will be injected as ΔP will either be zero or negative in terms of desired direction of flow of liquid refrigerant from the bleed line into the closed thread of the compressor.

As an example the ΔP at the injection port under minimum compression ratio conditions for **R22** at 40°/105°, would be 59 psid and at 20°/145°, referred to earlier, the differential would be 258 psid, determined as follows: The liquid refrigerant injection pressure must be an acceptable amount above injection port pressure to insure continued injection. The measured port pressure P_I follows the relationship (P min + (P max − P min)/3) the ΔP (liquid to port) is then ($P_D - P_I$). For example,

$$P_I = P\,\text{min} + \frac{(P\,\text{max} - P\,\text{min})}{3} = 1.5\ P_S + \frac{(3.0\ P_S - 1.5\ P_S)}{3}$$

$$= 1.5\ P_S + 0.5\ P_S = 2.0\ P_S$$

at 40°/105° $P_D - P_I = 225 - 2 \times 83 = 225 - 166 =$ 59 psid

at 20°/145° $P_D - P_I = 374 - 2 \times 58 = 374 - 116 =$ 258 psid

Due to the structure of the machine, it may be impractical to inject into the end plane of the compressor, that is, axially, the injection port P must normally be moved back within the bore at a corresponding helix angle of the rotor. Assuming in the present example that the wrap angle of the female rotor is 200°, moving the injection port P, FIG. **2**, back 0.400 inches from the discharge end, the optimum point of liquid injection which is equal to a wrap angle $\measuredangle$ of 215° from suction, which is established in the prior step must have added thereto the distance moved back the rotor bore times the wrap angle divided by the diameter of the rotor times the rotor ratio of *L/D*. assuming that the rotor diameter is 102 mm or approximately 4 inches, and a *L/D* ratio of one, the calculations are as follows:

$$\angle' = \angle + L' \left(\frac{\alpha_1}{D\ L/D}\right)$$

L' = 0.400 inches from discharge end
$\angle$ = 215°
α_1 = wrap angle = 200°
D = 102 mm = 4 inches
L/D = 1

therefore:

$$\angle' = 215 + 0.4 \times \frac{200}{4(1)} = 215\ 20° = 235°$$

In achieving the determination of the optimum point of injection, the match of the built-in pressure ratio to the actual operating pressure ratio is significant. When the actual compression ratio *C/R* is narrower in variation, there is more latitude in the location of the point of injection because on one end there is the concern with leak back to suction, and on the other end, there is the concern of overcompression prior to exposure of the discharge porting which locates the injection point further back towards the suction. The best brake horsepower per ton obviously occurs if there is absolute match between the discharge port location to the actual operating parameters of the compressor, and that the point of liquid injection is not earlier than need be.

What is claimed is:

1. A method of determining the optimum point of injection of liquid refrigerant into a compressor working space of a helical screw rotary compressor forming a component of a refrigeration system including a condenser coupled to the discharge side of said compressor, and wherein said compressor includes a housing forming a working space comprising two intersecting bores, a suction port at one end of said housing opening to said space for admitting gaseous refrigerant as the compressor working fluid thereto and a discharge port within said housing opening to said space for discharging compressed working fluid for passage to said condenser, two intermeshing helical screw rotors rotatably mounted in said bore for determining compressor capacity and a shiftable slide valve for returning a variable portion of the working fluid back to compressor suction prior to compression thereof, and wherein said compressor discharges a compressed refrigerant gas into the condenser of the refrigeration system and said system includes means for bleeding off a portion of said refrigerant in liquid form from said condenser and injects said bled liquid refrigerant via an injection port into one of said bores for cooling the gaseous working fluid during compression thereof, said method comprising the steps of:

1. determining the minimum built-in volume ratio of the compressor;
2. determining within which bore injection is to take place;
3. determining the minimum operating compression ratio and maximum operating compression ratio of the compressor;
4. determining the pressure versus wrap angle plot for the rotor of the compressor both selected from step (2);
5. superimposing the determined parameters from step (1) and step (3) on the plot of step (4); and
6. ascertaining from the plot of step (5) subsequent to step (4) an acceptable injection point for said injection port in terms of wrap angle from suction of the screw rotor for the bore as determined from step (2) which results in continuous injection of liquid refrigerant within said working space regardless of undercompression or overcompression of the working fluid and the position of the slide valve within the limits of the parameters determined by steps (1), (2) and (3), without significantly compromising compressor efficiency.

2. The method of determining the optimum point of injection of liquid refrigerant into the compressor working space of a helical screw rotary compressor, as claimed in claim **1,** wherein the injection port is radial and located at an axially displaced position relative to the discharge end of said bore and wherein the injection angle as determined by said method is corrected for wrap angle variation of the rotor carried by the selected bore due to said axial displacement of said port by means of the relationship:

$$I' = \angle I + L' \frac{\alpha_1}{D \times L/D}$$

1

where:

$\angle'$ = is the injection angle after compensation for rotor wrap angle due to axial displacement of said port in degrees
$\angle$ = uncompensated injection angle in degrees
L' = is the selected axial distance from the discharge end of the working space to the injection port in inches
α_1 = wrap angle of the screw rotor for the selected bore in degrees
D = diameter of the screw rotor in inches
L/D = ratio of rotor length to its diameter.

3. In a helical screw rotary compressor forming a component of a refrigeration system, said system including in order from the compressor, a condenser, a thermal expansion valve and an evaporator, and wherein said compressor includes: a housing forming a working space comprising two intersecting bores, a suction port at one end of said housing opening to said space for admitting gaseous refrigerant as the compressor working fluid to said space and a discharge port within said housing opening to said space for discharging said working fluid after compression, two intermeshing helical screw rotors rotatably mounted in said bores and defining with said bores working chambers wherein said compressor discharges compressed refrigerant gas into said condenser of the refrigeration system and a shiftable slide valve for returning a variable portion of the working fluid back to the suction port prior to compression, and said system includes means for bleeding off a portion of said refrigerant in liquid form from said condenser and an injection port opening up into said working space for injecting bled liquid refrigerant therein for cooling the working fluid during compression thereof, the improvement wherein: said liquid refrigerant injection point is located at a point relative to a given compressor working chamber determined by the instantaneous pressure relationship between the bled liquid refrigerant and the pressure of the compressor working fluid during compression within said chamber, such that cooling of the working fluid during compression is assured by continuous injection of bled liquid refrigerant at said point regardless of compressor slide valve location, and overcompression or undercompression of the working fluid due to changes in system condenser and evaporator loads and without significantly compromising compressor efficiency wherein said injection port is located within the bore housing said female screw rotor, said female screw rotor has a wrap angle of 200° and said port is located within the bore housing the female screw rotor at a wrap angle from suction within the range 210° to 226°.

4. The helical screw rotary compressor as claimed in claim **1**, wherein the position of the liquid refrigerant injection port in terms of the wrap angle of the female screw rotor from suction is 215°.

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