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The High-Ratio Circulating Compressor

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THE HIGH-RATIO CIRCULATING COMPRESSOR

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ABSTRACT

A high ratio, positive displacement, transverse flow, lobed rotor recirculating compressor is described herein. The design features an integral downstream heat exchanger and integral recirculation ducts that convey cooled discharge-pressure fluid back to intermediate ports.

The physical arrangement provides continuous recirculation flow back into the displacement cavities without concurrent communication with either discharge or inlet ports. Such communication is prevented by favorable geometry, including the use of 4-lobed or 5-lobed rotors.

Through optimized recirculation flow and minimized adiabatic compression, performance capability for this type of compressor is greatly expanded. Limitations associated with working fluid temperature rise and thermal distortion are removed. Reduced internal rotor clearances result in high volumetric efficiency.

INTRODUCTION

Ongoing photochemistry process research, conducted by the Los Alamos National Laboratory for the Department of Energy, has involved development and extensive modification of commercial Root's type compressors for use in an experimental gaseous flow system. The development effort was expanded from the initial scope to include evaluation and possible modification of recirculating compressors for use in a later phase of the program.

In carrying out the evaluation effort it soon became evident that only part of the potential performance capability had been realized in present state units. The high ratio compressor design was originated to develop the entire potential by fully exploiting thermal difference between pressure gain from recirculation flow and pressure gain from adiabatic compression.

PRESENT STATE

Current versions of recirculating compressors are available from a limited number of manufacturers in two or three different rotor configurations. When compared to the original Root's design they all have improved performance capability, but at varying levels. In general, they all benefit from a net reduction in temperature rise through the compression cycle.

This reduction is achieved by raising displacement cavity pressure with recirculation fluid before the cavity opens to discharge. This reduces the amount of pressure differential requiring adiabatic compression. Pressure gain from recirculation is subject to temperature rise from some degree of flow energy conversion, which is not exponential. It is based on a pressure-volume relationship, and tends to approach a level value with increasing pressure differential.

In all of these units, recirculation flow is either interrupted or reversed as a rotor lobe sector passes by the intermediate port.

The displacement cavity is not completely filled by the flow before it opens to discharge. This leaves a significant amount of pressure differential still subject to adiabatic compression. While seal material and bearing lubricant lifetimes are no longer troublesome, pressure ratios much greater than 4:1 generate overall temperature gains that limit the performance.

In summary, present day Root's type recirculating compressors are capable of improved performance over that of the original Root's design, particularly in the subatmospheric pressure region. Reduced temperature rise in the working fluid has permitted internal clearances to be tightened, thereby improving volumetric efficiency. However, a significant part of the cycle pressure differential is still subject to adiabatic compression, and thermal distortion in the compressor housing remains a problem at higher pressure ratios.

COMPRESSION CYCLE SEQUENCE

Figure 1 is a flow schematic of the recirculating compressor cycle. It is formulated to identify and account for all of the flow related occurrences taking place during the cycle. It provides a basis for math modeling, and is applicable to both present state and high ratio compressors.

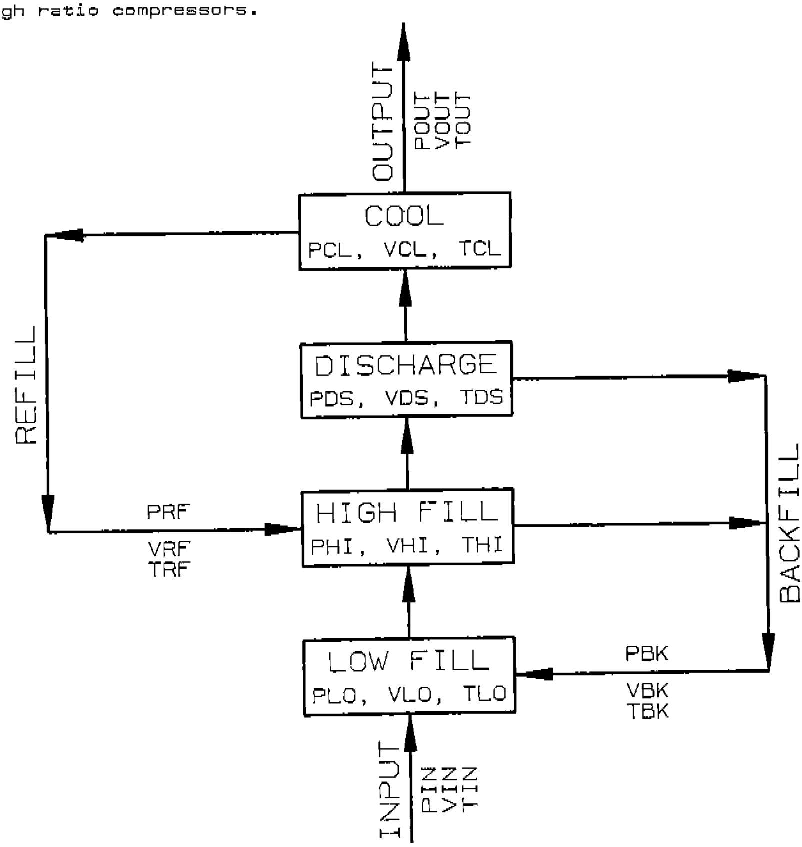

FIGURE 1

COMPRESSOR CYCLE FLOW SCHEMATIC

Figure 2 is a cross section of the recirculating compressor that identifies cycle features of the sequence described herein. The figure is based on high ratio design configuration for clarity.

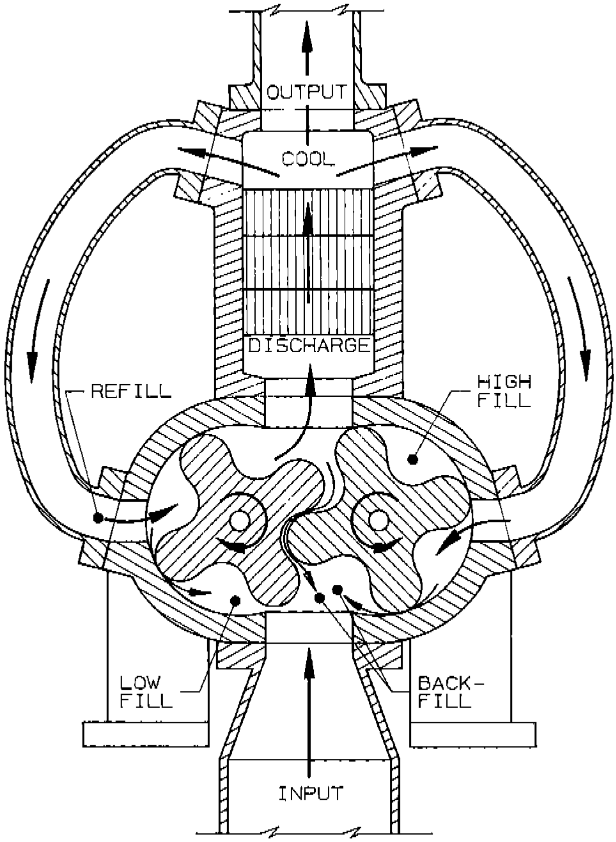

FIGURE 2
COMPRESSOR CROSS SECTION

Low Fill:

As a displacement cavity rotates through the intake region, it is filled primarily with fluid entering through the intake port. A secondary but significant source of fill consists of fluid feeding back from the discharge region through rotor-mesh clearances, and from the preceding cavity through rotor-housing clearances.

Temperature of the input fluid after it enters the cavity may be slightly higher due to flow-related effects. Pressure at low fill completion is slightly lower than upstream pressure due to entrance losses. Input temperature rise and fill completion factors should be included when math modeling.

Feedback through rotor mesh and rotor housing clearances consists of adiabatic expansion with friction through converging-diverging passages and through high L/Dh passages. Passage area for a particular compressor equals the clearance area. Above a critical pressure ratio of about 2:1 velocity through the passages is sonic, and backfill flow rates are calculated on that basis.

At low fill completion, fluid within the displacement cavity is a mixture of input fluid and feedback or backfill fluid. Time duration for the low fill and for the succeeding refill and discharge sequences is typically 5 to 10 milliseconds each.

High Fill:

After low fill completion the displacement cavity opens to the refill port. Cooled, high pressure fluid recirculated from downstream of the heat exchanger flows into the cavity. Refill pressure at the port is compressor output pressure less the recirculation loop pressure drop.

Pressure within the cavity at high fill completion depends on particular compressor characteristics. For present state commercial units, high fill pressure is significantly less than refill pressure, due largely to overlap between refill and discharge sequences.

In simple terms, the refill sequence is analogous to a tank filling operation. As long as the displacement cavity remains closed to discharge, the volume remains constant. There is no heat input or rejection, and no mechanical work in or out. Pressure increase results from a large influx of mass, accompanied by some degree of flow energy conversion.

However, flow dynamics in actual compressor operation differs significantly from the tank filling procedure. Refill passes through a large port into a small fixed volume in a very short time period of 2 to 10 milliseconds. Average flow velocity through the port is not greater than Mach 0.05. Average flow velocity of the fluid trapped within the cavity after filling is typically not greater than Mach 0.10.

Heat gain in refill fluid as it enters the cavity is calculated from a PV relationship, with P being the average pressure difference across the refill port and V being volume of the fluid prior to entering the cavity. A Refill Dynamics Factor, which is a multiplier less than unity, is used to account for the flow dynamics effect.

Flow energy conversion in recirculation flow as it enters the displacement cavity is a primary factor in determining thermal efficiency. In compressors based on the reference design where adiabatic compression is minimized, it becomes the major source of heat exchanger load.

Discharge:

After high fill completion the cavity opens into the discharge region. The fluid is adiabatically compressed from high fill pressure to discharge pressure, which is higher than output pressure by the amount of heat exchanger pressure drop. Total compression required includes pressure drop through the heat exchanger and recirculation ducts, plus the refill entrance loss.

The cavity, in returning to the intake region, is progressively swept by the mating rotor lobe to hold back compressed fluid. Because of the pressure difference, significant feedback into the intake region occurs through clearance between rotors as they rotate through mesh.

Cool:

From the discharge region fluid going to output and to the recirculation loops passes through a heat exchanger to lower the temperature to the output state. Mass flow through the heat exchanger is that at discharge less the backfill mass flow. Mass flow to output is equal to the input mass flow.

THE HIGH RATIO REFERENCE DESIGN

Figure 3 shows a cross section of the High Ratio Reference Design arrangement.

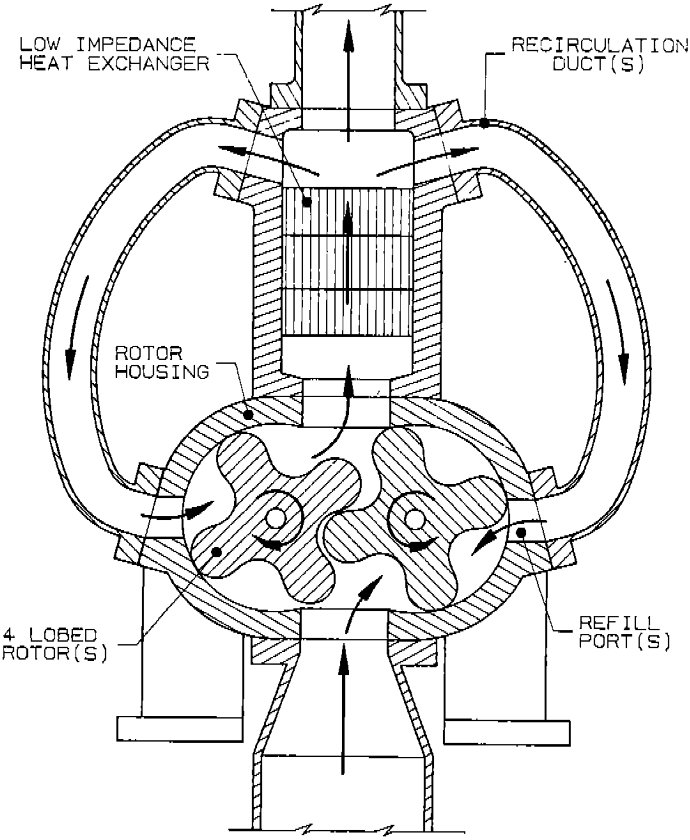

FIGURE 3
HIGH RATIO DESIGN FEATURES

In recirculating lobed rotor compressors of this type, further expansion of performance capability can be achieved by minimizing adiabatic compression and optimizing recirculation flow input. These two mutually supporting measures essentially remove performance limitations associated with working fluid temperature rise and thermal distortion.

Because of inherent cycle sequence overlap and/or recirculation flow blockage, 2 or 3 lobe rotor configurations are not capable of high ratio compression performance. A 4-lobe configuration permits excellent recirculation flow continuity and low impedance porting. Rotor configurations having more than 4 lobes sacrifice volumetric displacement, but may prove useful where very low impedance porting is a determining consideration.

To fully develop the potential performance capability, a high ratio reference design has been established featuring a 4-lobe rotor. The design effectively eliminates cycle step overlap and recirculation flow disturbances that characterize the present state compressors. The design also features both an integrated heat exchanger and low impedance recirculation flow loops. These carry cooled output state fluid back to the intermediate or refill ports without significant pressure loss.

The displacement cavity closes to refill before it opens to discharge, and cavity pressure from recirculation input rises to from 94 to 97% of discharge pressure. Adiabatic compression is thereby held to a minimum, and pressure-pulse perturbation into discharge flow has essentially been eliminated.

<u>Rotor Geometry:</u>

To develop continuous mesh contact between rotors throughout full rotation, the following lobed rotor geometry has been incorporated into the high ratio reference design.

N = the number of rotor lobes.
Pitch diameter (p.d.) = rotor center distance.
Pressure angle (p.a.) = 360/4N (normal lobe).
Base circle diameter (b.c.d.) = p.d. * cosine p.a.
Lobe tip/root radius (r.t.r.) = (pi * b.c.d.)/4N.
Rotor outside diameter = p.d. + (2 * r.t.r.).
Rotor root diameter = p.d. - (2 * r.t.r.).
Root and tip radii are centered on the pitch diameter.
Lobe form is involute between root and tip radii.

<u>Backfill or Slippage:</u>

The method of calculating backfill or slippage is applicable for high ratio pressure differentials, where mass flow through converging-diverging passages is limited by sonic choking at the throat, and where mass flow through high L/Dh ratio passages is limited by sonic choking at the exit. For simplicity these passages are lumped together and called "rotor clearance area". They include rotor-to-rotor mesh clearance, rotor tip-to-housing clearances, and rotor end-to-housing clearances.

Rotor-to-rotor mesh clearance configuration is primarily a converging-diverging passage (see Figure 4a). Whenever a rotor lobe tip passes through the mating rotor root the configuration abruptly changes to a high L/Dh ratio passage (see Figure 4b), and then abruptly back to the converging-diverging passage. For the high ratio reference design the configuration switch occurs every 2 to 4 milliseconds. Pressure differential across the passage is constant.

Rotor tip-to-housing configuration forms a uniform converging-diverging flow passage (see Figure 4a). Pressure differential across the passage(s) is variable, starting from zero at intake closure and reaching compression ratio level by the time the lobe tip reaches the recirculation port.

Rotor end-to-housing configuration forms a uniform high L/Dh ratio flow passage (see Figure 4b). Pressure differential across the passage is variable, subject to essentially the same variation that the rotor tip-to-housing passage encounters.

Figures 4a and 4b show backfill flow passage configuration and typical reduced clearance values.

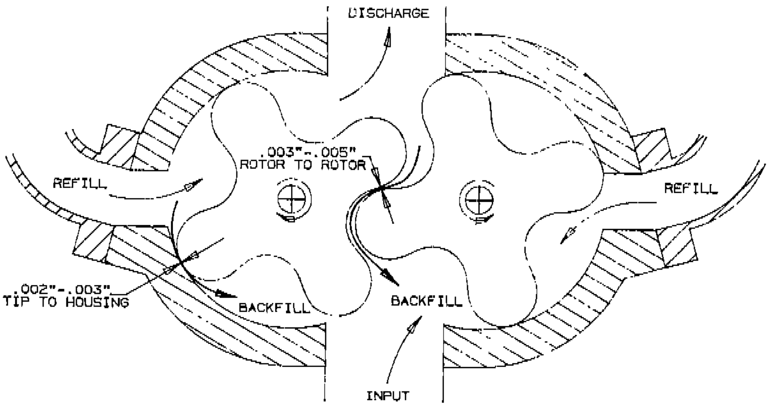

FIGURE 4a

CONVERGING-DIVERGING PASSAGES

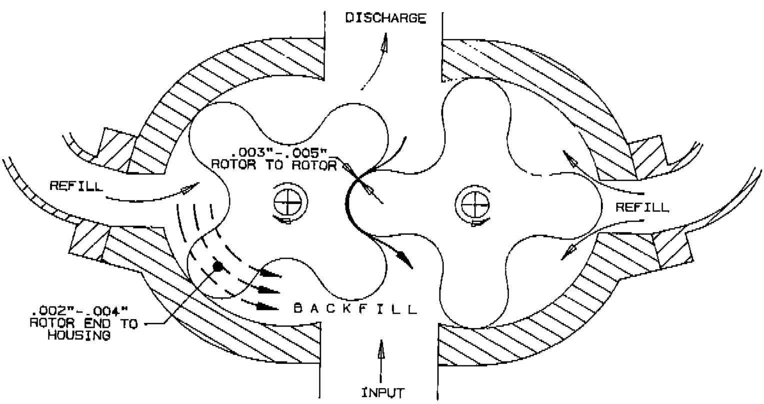

FIGURE 4b

HIGH L/Dh RATIO PASSAGES

The calculation for backfill volume requires the equivalent of a nozzle discharge coefficient, a multiplication factor less than unity. This Backfill Discharge Factor accounts for surface velocity effects from rotor motion, and adjusts for velocity profile across the passage width. Average flow velocity through very narrow passages is significantly less than sonic.

Volumetric Efficiency:

For recirculating compressor units based on the high ratio reference design, calculated volumetric efficiency is greater than 90% for compression ratios up to 5:1, and greater than 80% for compression ratios up to 10:1. By contrast, volumetric efficiency for present state recirculating units drops to a level where economic operation is no longer feasible beyond a compression ratio of 4:1.

Development and Design:

Although development of this high ratio compressor is still in an early state, design requirements and features are fully supported by existing technology. Present state materials, machine tools, fabrication-assembly methods, and drive train components are used throughout. The associated development effort required is neither complex nor extensive.

Design is based on present rotating machinery engineering practices, and is also quite straightforward. The rotor geometry creates no dynamic balance difficulty and is torsionally very rigid. Acoustic intensity at inlet is not expected to be troublesome, but may require treatment for particular applications. In general, the design is inherently free from sources of significant dynamic and acoustic disturbance.

SUMMARY

The high ratio recirculating compressor design is based on the Root's positive displacement rotary blower that has been serving industry since 1854. Like the Root's blower it is simple, rugged and trouble free. It has a large capacity for a given physical size. It has no valves, and no reciprocating or rubbing parts. Toxic and aggressive gases and vapors can be handled as well as normal gases. With proper shaft sealing there is no contamination of the working fluid.

Unlike the Root's blower, it has no performance limitations originating from temperature rise in the working fluid. As a compressor, single stage performance capability includes 10:1 ratio compression with good volumetric efficiency. As an evecuator, single stage ultimate pumpdown to atmospheric pressure is less than 30 torr.

When fully developed, the high ratio recirculating compressor is expected to gain widespread usage in commerce and industry, both as a compressor and as a vacuum pump. It can handle the more difficult fluid moving requirements, and is uniquely qualified for service where temperature level and contamination in the working fluid are matters of concern.

The units can find many applications in the chemical process industry, and can serve to drive gaseous laser flow loops. They can also be used for many applications now served by sliding vane, rotary screw, or liquid ring compressors. In time, the high ratio recirculating compressor can become one of the most versatile of the positive displacement types.