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The CONTROLLED ROTARY VANE

GAS-HANDLING MACHINE

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ABSTRACT

The design, development, test and application of an advanced rotary gas-handling device, termed a Controlled Rotary Vane (CRV) machine is reported. This machine is related to conventional sliding vane compressors except that the motion of the vanes of the CRV is controlled. This control is such that the vane tips closely approach but do not contact the interior of the stator contour; the visco-inertial effect of the circulating lubricant aids in sealing.

Stator contour-independent vane motion control in the CRV machine not only results in a significant reduction in direct mechanical friction but, as important, allows low-loss internal fluid dynamic design for further increases in overall gas-handling efficiency. This design flexibility is manifested as generous porting, optimized volumetric change/rotation derivatives and effective internal peripheral sealing across the inlet and output ports. Such compressor design yields an efficient and compact machine suitable for operating with low density gases. This machine operates on such refrigerants as low pressure aerosol spray can propellents. These propellants are approved both by the U.S. Environmental Protection Agency and the U.S. Consumer Product Safety Commission as replacements for CFC propellants that were banned in the U.S. and several other countries in 1978.

Laboratory and field testing of this type of machine has empirically demonstrated significant performance enhancements resulting from de-coupling vane motion from the stator contour. Independent tests conducted on a 30 cid (490 cc) CRV compressor operating as a refrigerant compressor using CFC-114 demonstrated a COP of 4.67 while operating across a standard test condition of 45F/120F (7C/49C). This performance reflects an overall isentropic efficiency closely approaching 80% and a volumetric efficiency of nearly 90% while producing about 40,000 BTUH (10,000 kcal/hr) of cooling.

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I. INTRODUCTION

THIS PAPER REPORTS on an efficient, large displacement, small envelope size gas-handling machine that has proven suitable for use in air conditioning and small-scale Rankine power generation using low vapor-pressure fluorocarbon as well as hydrocarbon refrigerants/working fluids. This gas-handling device, an example of which appears in Figure 1, is termed a Controlled Rotary Vane (CRV) machine and consists of four basic parts: a stator housing, a set of two mirror-image stator cam plates, a rotor and a set of roller-controlled radial vanes. Both ends of these vanes are equipped with fixed axles and bearing rollers that follow the continuous internal cam surfaces built into each of the two stator cam plates which form the ends of the CRV machine.

The vane assembly functions within the CRV in such a way that internal rubbing friction is virtually eliminated. This is because the tips of the vanes maintain a small clearance from the inside of the substantially elliptical contour of the interior of the CRV stator housing. This occurs as the axle-roller assemblies fixed to the vanes follow the internal cam paths built into the stator end plates. In addition, low-loss internal gas dynamic design, discussed subsequently, provides for further gains in machine efficiency. Internal sealing is maintained by the visco-inertial effects of the lubricant within the machine.

A variety of low vapor-pressure refrigerants have been tested in various CRV systems. These include such hydrocarbons as isopentane, isoamylene, neopentane, n-butane and Phillips A-17 aerosol spray can propellant. In addition, the machine has operated well with controlled substance fluorocarbons CFC-11 and CFC-114. Laboratory tests showed that compressor coefficients of performance were basically the same for both types of refrigerants, but non-fluorocarbon COP's were slightly higher than the fluorocarbon COP's (1). Performance tests of a 30 cid (490 cc) R-114 refrigerant CRV compressor conducted by a major independent testing firm (2) demonstrated a cooling coefficient of performance of 4.67 ("EER = 15.9) with CFC-114 at a standard rating condition of 120F condensing and 45F evaporation.

The CRV machine, in various configurations and displacements, has been successfully demonstrated with numerous low pressure refrigerants in the following applications:

1. Non-CFC automotive air conditioning

2. Air-to-air residential air conditioning

3. 1-2 kilowatt solar/thermal electric power generation in the organic Rankine cycle

4. Water-to-water and water-to-air reversing heat pumps

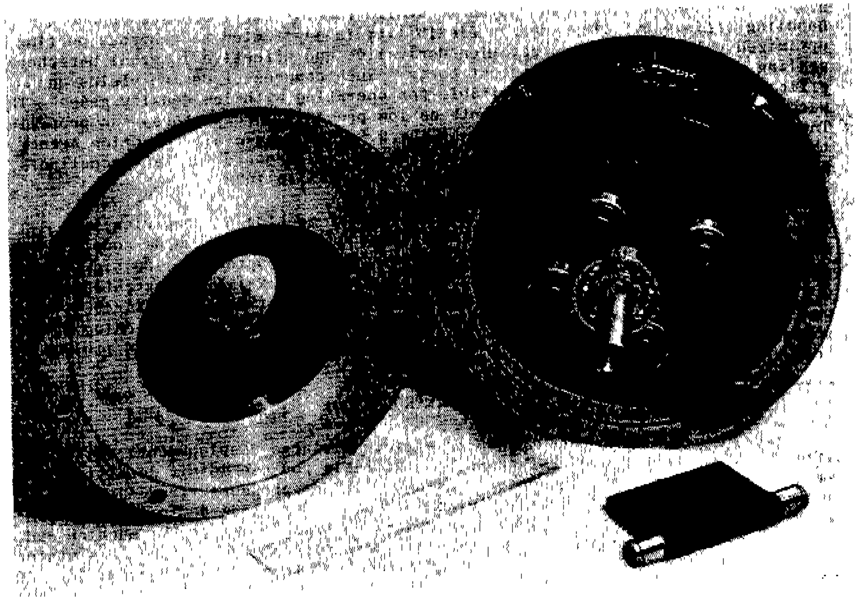

Figure 1: Photo of the Model 230 CRV with End Plate Removed

II. BACKGROUND: The Conventional Rotary Vane Machine

Conventional rotary sliding vane compressors have long been known and commercially utilized due to their variety of positive attributes. These include mechanical simplicity, smoothness of operation, small relative size and ease of manufacture. In spite of these advantages, however, the usage of such machines has fallen short of desired potential, partly due to relatively low efficiency.

The basic reason conventional sliding vane machines are inherently limited in performance is rooted in their attractive but simplistic mechanical design wherein the vane tips rub the internal contour of the compressor stator. Such design strongly inhibits port cross sectional flow areas. This results, of course, in significant fluid-mechanical flow losses, especially at higher levels of mass throughput. Further, the vane tip rubbing friction, along with increased vane/slot friction induced by the tip friction vector, contributes additionally to the basic efficiency limitations of the conventional sliding vane compressor. Thus, the purpose of a well-designed CRV machine is to maintain the basic charm and positive attributes of the conventional rotary vane machine but exclude its attendant negative feature of limited performance, but to do so at an effective cost. Briefly, high

Briefly, high performance is achieved by optimizing the fluid-mechanical properties of the machine while concurrently minimizing mechanical friction through the application of a simple static cam arrangement. This machine optimization results directly from controlling vane motion, not via the usual vane tip-contact method, but instead by contraining this motion by an independent cam guided means, as indicated above. The following Section is directed towards considerations related to the design of the CRV machine to accomplish this optimization. While the CRV machine will be discussed primarily as a compressor, the design factors also relate to the machine when operating as an expander.

III. GENERAL CRV MACHINE DESIGN

Basic Considerations

Recall that, fundamentally, the function of any compressor can be considered as threefold: a) inducing gas into itself, b) compressing the gas, and c) discharging the gas. In order for the compressor to be efficient, it is, of course, necessary that all three of these processes be efficient. From the stand point of the fluid mechanics of a positive displacement machine, one must minimize velocity changes (or losses associated with them) of the gas flowing into and out of the machine to maximize flow performance. As well, all three of these flow processes must be attended by a minimum of mechanical friction.

In a real positive displacement compressor, for example, high inlet efficiency is obtained by providing two favorable fluid-mechanical conditions: 1) large inlet port flow cross-sectional areas and 2) completion of the inlet volume expansion well before the inlet port closes. More concisely, this second factor can be expressed as:

$$dV/d\phi \mid_{\text{port closing}} = 0$$

where $dV/d\phi$ is the rate of change of inlet volume of the machine as a function of ϕ, an independent variable such as time or rotation angle

In the event that a flow velocity change occurs during the compression process itself (as in the case of rotary machines), the ideal rate of change of displacement volume during the discharge process would be constant. That is:

$$dV/d\phi \mid_{\text{discharge}} = \text{constant}$$

Combining a constant $dV/d\phi$ during discharge with generous discharge port areas, of course, minimizes fluid flow losses during the compressor's discharge process. As discussed and illustrated below, the CRV compressor can be designed to closely approximate these design ideals while yet retaining the many positive attributes of the well-known rotary sliding vane compressor.

CRV Machine Geometry

A useful set of geometrical relationships can arise between an ellipse, which substantially represents the inner contour of the CRV stator, and a circle, whose axis is off-set from the ellipse and which represents the rotor, that are relevant to the design of the CRV machine. These simple geometrical relationships provide for three important machine functions, all while maintaining only a single in-and-out stroke of each vane per rotor revolution. These functions are: 1) a natural inlet port closure wherein the inlet volume change has ceased well before actual port closure, 2) a gradual compression process and 3) a gradual constant rate of volume change during the discharge process. Further, all this is achieved while creating a long and effective sealing path between the high and low-side of the machine. This seal path is created in the vicinity of the point of tangency between the offset circle, which represents the rotor, and the ellipse, which represents the interior of the stator wall. These relationships are shown in Figure 2 In accordance with one of the important aspects of the CRV machine design, as just indicated, this rotor double-offset from the stator center can be chosen in such a way as to insure a single vane reciprocation stroke with each rotor revolution. While there are several obvious advantages to such a geometry, the most important is insuring mechanical friction minimization which is achieved in general by minimizing physical sliding displacement of the CRV machine parts causing volumetric changes.

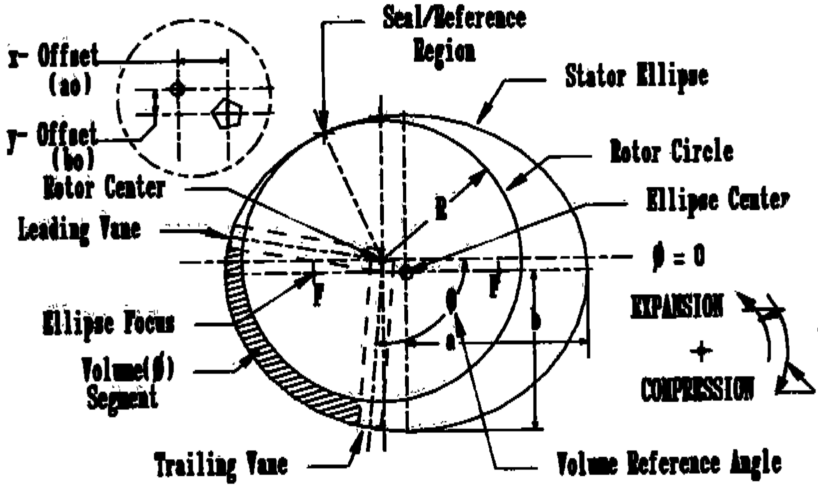

FIGURE 2: Stator/Rotor Geometry of the CRV Machine

IV. The MODEL 230 ROVAC CONTROLLED ROTARY VANE MACHINE

Figure 3 is a partial front view machine drawing of a Model 230 CRV, which is a third-generation successor to the Model 227 CRV compressor, whose performance was independently evaluated (1). The 230 CRV machine is designed for several applications including residential air conditioning using, for example, CFC-114, neopentane or A-17 spray can propellent. This four-vane machine has a swept volumetric displacement of about 26 cubic inches (420 cc) per revolution and produces approximately three tons of cooling at residential air conditioning operating conditions. It's 20-year design life speed is 1800 rpm and is 7.25 inches (184 mm) in diameter and 4.00 inches (100 mm) in main body length.

Summary Design Sequence

After having determined the approximate required volumetric displacement for a CRV, the next step involves the determination of a variety of geometrical relations, the first being the major axis and eccentricity of the stator ellipse. The model 230 CRV compressor design required a swept volumetric displacement of about 26 cubic inches (420 cc) per revolution and the external stator diameter and overall body length were limited to about 7.25 inches (184 mm) and 4.5 inches (115 mm), respectively.

With these constraints established, the smallest possible stator ellipse eccentricity value (defined here as the arc cosine of the ratio of the minor-to-major axes of the ellipse) is generally chosen. This is done in order to minimize the acceleration loads imposed upon the sliding vanes and their companion axle-roller assemblies. Further, such design maximizes the space available for the vane axle roller assemblies within the central core of the machine. In the case of the Model 230, the stator ellipse major axis was chosen to be 5.8750 inches (150 mm) with an eccentricity of 25.276 angular degrees. These choices permitted an active rotor length of 2.5 inches (63.5 mm), a rotor shaft diameter of 0.875 inches (22.225 mm) and vane roller diameter of 1.0625 inches (27 mm) without undue congestion in the central portion of the machine. Of course, the largest possible roller diameter is chosen to minimize the angular speed of these rollers in order to maximize their life and reliability.

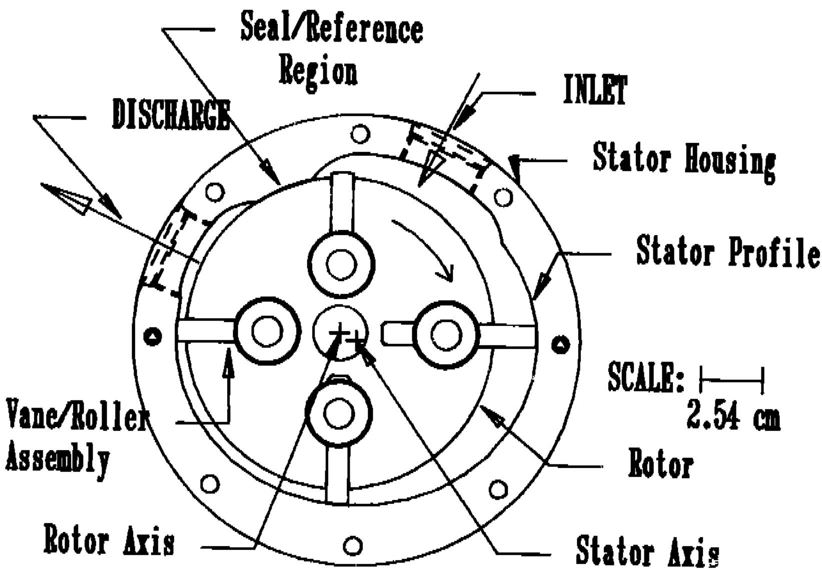

FIGURE 3: Frontal Assembly Drawing of the ROVAC Model 230 CRV Machine

Having specified the ellipse geometry, the next design step is to define the diameter of the rotor and it's x- and y- offsets from the stator ellipse. As a general rule, the diameter of the rotor should be as large as possible in concert with the desired volumetric displacement. This is because the ratios of in-slot vane length to the exposed vane "overhang" will be largest and therefore the induced vane/vane slot friction vectors will be minimized. For the Model 230, the rotor diameter was chosen to be 5.000 inches (127 mm) and the x- and y- offsets, ao and bo, were eventually specified as 0.2708 inches (6.88 mm) and 0.1180 (3.00 mm), respectively. The angle of the rotor/stator seal region was chosen to be about 25 degrees. These geometrical selections, along with four vanes, produced the desired volume ratio of approximately three-and-a-half to one.

The number of vanes should be the minimum required to obtain the desired pressure/volume ratio and still maintain vibration-free operation. Further, some control over actual machine volume ratio is presented by the actual location of the opening of the discharge port. For example, delaying the port location (and thus it's opening position) increases the volume ratio. However, this design latitude is relatively small because, as discussed subsequently, delaying the discharge point opening too much will not only result in poor flow area, but the desired constant rate of discharge volume change will be partially sacrificed.

General CRV Design Considerations

In relatively low pressure applications, say about 100 psia (700 kPascals), the majority of the present CRV machine clearances and tolerances are on the order several thousandths of an inch (about 0.05 mm) and the bearing tolerances are ordinary "class-three's". All CRV machines to date use standard refrigerant oils for lubrication.

It is interesting to speculate that the controlled rotary vane machine can be made in sizes not previously considered for rotary vane-type machines; vane tip speed is limiting in conventional rotary vane machine due both to loads and rubbing speeds. Obviously, the CRV compressor has no such specific limitations. On the other hand, the CRV compressor does not appear, on the surface, at least, to be readily adapted to sizes much smaller than, say, one-half ton of refrigeration or a volumetric displacement on the order of 5 cid (80 cc) because of machine congestion potentially offered by the axle-roller assemblies of the controlled vanes.

When the CRV machine is brought to speed, centrifugal forces acting upon the vane, as well as internal vane heel pressure forces, bring the rollers into rolling engagement with the internal cam profile in the end plate. Due to the machine geometry just discussed, the vane tips do not actually touch the interior of the stator profile. Instead, the vane tip becomes tangent at some angle-specific location to the stator profile at a clearance distance on the order of 0.001 - 0.002 inches (0.025 - 0.050 mm). The point of clearance tangency varies with the angular location of the vane and the cam contour design must account for this variable location. Due to the relatively large number of geometric factors that must be accounted for, the contour profile of the cam path is machine computed.

As discussed initially, the most significant design feature of the CRV is its rotating vanes which are cam/roller-controlled so that the tips of these vanes do not actually engage the internal wall of the stator housing; clearing by several thousandths of an inch. Effective gas sealing across this clearance is achieved through the viscous and inertial properties of the lubricants which are conventional refrigeration oils in the 150 - 300 SUS viscosity. In addition, such geometrical considerations such as liberal vane tip radii and vane widths enhance internal gas sealing. Although not thoroughly investigated at time of writing, preliminary and informal CRV testing indicates that "rough" (on the order of 125 rms finish or larger) vane tips and interior stator walls contribute to better internal gas sealing; presumably due to enhanced lubricant cling.

Model 230 CRV Compressor Performance

While considerable test performance information has been generated during the development of the CRV machine technology, one particular set of performance data will be highlighted because it was generated independently by the Electrical Test Laboratory (ETL), of Cortland, New York (1). These performance tests were conducted on a Model 227 6-vane 30 cid (490 cc) CRV machine (the predecessor to the Model 230) operating as a vapor refrigerant compressor in order to determine the Coefficient of Performance (COP) of the machine when used as an air conditioning or heat pump compressor. These independent performance tests were conducted in accordance with applicable sections of ASHRAE Standard 23-78 and Section 5.1 of ARI Standard 520-78. Both test standards apply to positive displacement compressors.

The Model 227 compressor tests summarized in the ETL Report were conducted with refrigerant CFC-114 with the CRV compressor operating at 1750 rpm across a standard operating condition of 45F (7 C) evaporation and 120F (49 C) condensation. Under these test conditions, a COP of 4.67 ("EER" = 15.9 BTU/Watt) was demonstrated. The four-vane Model 230 CRV, operating under the same conditions, demonstrated a COP of 4.85 ("EER" = 16.5). The adiabatic/isentropic efficiency of the Model 227 CRV compressor, was found to be 77%. The volumetric efficiency of this machine is 87%. The slightly improved Model 230 CRV demonstrated compressor adiabatic/isentropic efficiency of 80% and a volumetric efficiency of 92%.

V. DEMONSTRATED CRV MACHINE APPLICATIONS

As indicated earlier, CRV machines have been demonstrated in a number of applications under both laboratory and actual field conditions. These individual demonstration applications are briefly discussed below.

Automotive Air Conditioning

During the period from mid-1980 through the summer of 1981, an extensive non-fluorocarbon (non-CFC) automotive air conditioning system development program was undertaken in Florida. The objective of this program was to analyze, design, fabricate and test a non-fluorocarbon application of the Model 227 CRV in an automotive air conditioning configuration. This developmental automotive system used a low vapor pressure hydrocarbon refrigerant (isoamylene). The basic requirement of this program was the generation of comparative performance data between the installed developmental low pressure non-CFC development system and a well-developed conventional CFC-12 fluorocarbon automotive air conditioning system. To meet this requirement, twin vehicles were required; one equipped with the developmental CRV-based unit and the control vehicle equipped with the conventional standard CFC-12 automotive air conditioning system.

In order to secure the performance data, in addition to appreciable laboratory calorimeter testing, extensive car road testing was required. The road tests spanned a three-month Florida summer season during which time over two hundred operating hours and nearly 10,000 road miles were accumulated on the test car. The road tests involved two standard test patterns established by an automobile manufacturer. Pattern "A" monitored car cool-down performance from a thermally soaked condition at 55 mph (88 kph) for 30 minutes followed by a 30-minute idle test. Test pattern "B" monitored cool-down also from a thermally-soaked condition for a 30-minute period at 25 mph (40 kph). It was determined during these tests that the non-CFC test car using the Model 227 CRV compressor provided performance superior to the CFC-12 unit under both test pattern conditions. In addition, the fuel mileage of the CRV-equipped auto was consistently greater than the control vehicle by approximately one mile per gallon. Also, the temperature of the test car's engine coolant entering the radiator consistently remained between 5 and 10 Fahrenheit degrees (3 and 5.5 Celsius degrees) cooler than the control vehicle's. Reference (3) presents a detailed report on this non-CFC automobile air conditioning development program.

Residential Air Conditioning

In late 1982, a ROVAC 230 CRV was developed for use with a nominal 3-ton central residential air conditioning system using CFC-114 or a light hydrocarbon as the refrigerant. This split system was installed in a home in the Orlando, Florida, area, and has operated essentially without maintenance since installation.

A slightly improved prototype version of the original ROVAC central air conditioning system was evaluated in the test laboratory of a manufacturer of conventional residential air conditioning systems in July, 1983. Although a formal test report was not issued, these performance tests showed a system EER performance level slightly over 11 BTU/Watt.

Small-Scale Solar/Thermal Power Generation

In March, 1985, the first application of the CRV machine in the generation of electric power from low source and sink temperature differences was demonstrated in the organic Rankine cycle (ORC). This prototype power generation unit produced approximately 2 kWe of net electrical power while operating across boiling and condensation temperatures on the order of 100 Fahrenheit degrees (56 Celsius degrees).

Due to the promising performance of this prototype small-scale ORC power system, the device was developed into a commercially-available unit known as the ROVAC ORC-

2000 which is rated at a nominal 2 kWe capacity and can produce power with source temperatures obtainable by flat plate solar collectors as well as through the combustion of a very wide variety of low-grade renewable energy resources.

Water Source Reversing Heat Pump

In late 1985, the development of a water-source reversing heat pump was begun. A number of these demonstration units are currently operating satisfactorily New England field and laboratory environments. These systems, operating on several low pressure refrigerants (including non-CFC aerosol spray propellants) are showing competitive performance in both the heating and cooling modes of operation.

VI. CLOSURE: DISCUSSION and RECOMMENDATIONS

An appreciable body of laboratory and field experience with the controlled rotary vane machine has been generated that demonstrates a variety of successful applications of this device. While the present CRV designs already provide exceptional performance, it seems reasonable to assume that continued development of this type of machine will lead to further advances in efficiency and application. In this regard, a specific recommendation is to generate a precision computer model of the CRV that would lead to optimized geometries and performance for various capacities, sizes, refrigerants, and applications.

Because the CRV is able to operate efficiently with Environmental Protection Agency- and Consumer Product Safety Commission- approved low vapor pressure aerosol spray can propellant refrigerants, there now exists an alternative to the use of CFC-12, for example, in automotive applications. This is an important consideration due to the world-wide realization that fluorocarbons, while possessing acceptable thermodynamic properties for use in conventional high pressure compressors, have proven to be environmentally unfit -- on a global scale -- due both to upper stratospheric ozone depletion and Earth warming caused by these substances. Therefore, it is additionally recommended that continued development efforts be expended towards further refinement of non-CFC automotive air conditioning systems based upon safe and accepted low vapor pressure hydrocarbons.

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