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APPLICATION OF THE PERIPHERAL COMPRESSOR
TO THE FLUORIDE VOLATILITY PROCESS

UNION CARBIDE CORPORATION
NUCLEAR DIVISION
OAK RIDGE GASEOUS DIFFUSION PLANT

operated for the **ATOMIC ENERGY COMMISSION** *under* **U. S. GOVERNMENT** Contract W-7405 eng 26

**UNION
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APPLICATION OF THE PERIPHERAL COMPRESSOR
TO THE FLUORIDE VOLATILITY PROCESS

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A B S T R A C T

During the past several years, the U. S. Atomic Energy Commission has sponsored development programs designed to further basic technology relating to a fluoride volatility process for the recovery of valuable uranium and plutonium from spent nuclear reactor fuels. The ORGDP Gaseous Diffusion Development Division has been engaged in a variety of engineering and conceptual plant studies, as well as a component development program, in support of the volatility project. To lower operating costs in reprocessing plants by reducing reagent usage, gas recycle was proposed for use in the fluorination and hydrochlorination steps of the process. This report presents a detailed description of the development and testing of a prototype two-stage peripheral compressor designed for recycle service. An account is given of the maintenance requirements and operating problems encountered in over 2 years of intensive testing, including 6,200 hours of operation using mixtures of nitrogen, fluorine, and uranium hexafluoride. The overall performance and acceptability of the compressor, as it might be used in a 1-tonne of uranium per day reprocessing plant, were critically evaluated and found to exceed design criteria.

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APPLICATION OF THE PERIPHERAL COMPRESSOR
TO THE FLUORIDE VOLATILITY PROCESS

During the past several years, the Division of Reactor Development and Technology, United States Atomic Energy Commission, has sponsored a variety of economic and technical feasibility studies of processes for recovery of fissionable material from irradiated nuclear fuels. As a part of this effort, the Oak Ridge Gaseous Diffusion Plant has been engaged in the development of basic technology for treating both high and low enrichment fuels by volatility or dry gas techniques. The work has been performed in conjunction with several national laboratory programs, particularly those at the Argonne National Laboratory and the Oak Ridge National Laboratory. Conceptual plant studies, including economic analyses, and component development and semiworks testing projects were initiated at the ORGDP to support and supplement the national laboratory efforts.

In essence, the fluoride volatility process for recovering uranium and plutonium from irradiated zircaloy-clad uranium dioxide fuels consists of a decladding step, using hydrogen chloride gas to remove the zircaloy, followed, first, by oxidation to disintegrate the uranium dioxide pellets to a more reactive U_3O_8 powder and, then, by one or more fluorination steps to volatilize the uranium and plutonium as the hexafluorides. The chemical reactions are carried out in fluid-bed reactors.

To reduce reagent usage and thereby lower operating costs in fluoride volatility reprocessing plants, reactant gas recycle appears to be preferred over once-through schemes for both the hydrochlorination and fluorination steps of the process. The recycle compressor requirements envisioned for a 1-tonne of uranium per day reprocessing facility are summarized in table I. It can be seen that compression ratios of about 2 to 1 and flow rates as high as 150 scfm are needed. Additionally, the nature of the process demands that the overall design of the compressors be highly reliable and troublefree, with remote or semidirect maintenance feasible.

As a part of the ORGDP fuel reprocessing program, a diaphragm type compressor equipped with a remote head was purchased and evaluated to see if such a machine would be suitable as a recycle compressor for the proposed processes. These tests showed the diaphragm compressor to have several undesirable and questionable features which could restrict its application in a remote facility*. Specifically, these are: (1) amplitudes of vibration are inherently high, requiring use of flexible connectors; (2) periodic replacement of diaphragms is necessary; and (3) flow rates are easily affected by malfunctioning of the check valves and by

* Smiley, S.H., Brater, D. C., and Pashley, J. H., ORGDP Fuel Reprocessing Studies, Summary Progress Report, Fiscal Year 1964 through Fiscal Year 1965, Union Carbide Corporation, Nuclear Division, Oak Ridge Gaseous Diffusion Plant, October 29, 1965 (K-1649).

accumulations of small amounts of gas in the hydraulic fluids. In addition, flow rates for diaphragm machines are basically low, and relatively large and expensive units would be required to meet the target flow rates.

TABLE I
DESIGN CRITERIA
FLUID-BED VOLATILITY LOOP COMPRESSORS
1 Tonne of Uranium Per Day Line

	<u>Compressor No. 1</u>	<u>Compressor No. 2</u>
Suction Pressure, psia	14.7	14.7
Pressure Ratio	2.0	2.0
Volume Flow, scfm*	50 to 150	25 to 50
Suction Temperature, °F	-40 to +80	200
Process Gas Constituents	HCL, H ₂ , N ₂	F ₂ , N ₂ , UF ₆
Approximate Range of Molecular Weights	18 to 35	28 to 61
Maximum Inleakage, scfm	0.2	0.2

* Standard conditions are 32°F and 14.7 psia.

From the standpoint of fabrication and maintenance costs; predicted operational characteristics; and the speed, impeller size, and number of stages required, the peripheral compressor appeared to be a more suitable type of machine for the intended applications than either a diaphragm or a centrifugal unit. This report summarizes the development and testing of the experimental volatility process peripheral compressor and presents an overall qualitative evaluation of the unit's suitability as a recycle compressor in a nuclear fuel reprocessing plant.

SUMMARY

The developed compressor design consists of a peripheral impeller and a motor rotor mounted on a single shaft supported in grease-packed precision ball bearings. A nitrogen-buffered bushing seal isolates the bearings and motor from the process gas with a small amount of nitrogen inleakage.

Motor and bearing cooling is accomplished by a water jacket around the motor stator and an air thermal barrier between the compressor passages and the load bearing housing. In order to reduce the overhang moment on the shaft and to improve the rotor dynamic characteristics, two stages of compression are mounted on a single wheel with an interstage labyrinth type seal. Test data indicated that, through the use of removable inserts to change the channel dimensions, the requirements for both recycle applications could be met with one basic impeller and housing design.

Performance data from a single-stage experimental peripheral compressor demonstrated that a two-stage machine with a 10-inch-diameter impeller operating at 11,000 rpm with interstage cooling could be adapted to meet the flow and pressure requirements of both the hydrogen chloride and fluorine recycle applications. A development program was initiated to define optimum channel and impeller geometries and to determine reliable shaft seal and bearing designs. To assist in this work, a test rig was built to confirm the performance of the integrated compressor design, using air.

Following the successful performance of the test rig, a prototype peripheral compressor was built and installed in a specially constructed test loop so that the integrated unit could be dynamically evaluated under simulated volatility process conditions. The specific goals of the prototype testing program were to confirm the predicted performance characteristics of the compressor, to determine the reliability of the bearing and seal designs, and to establish maintenance and service requirements.

The performance of the prototype compressor exceeded design criteria set forth for the fluorination step of the process. Although the machine was not specifically tested under the conditions established for the hydrochlorination step of the process, there is no doubt that the compressor could adequately handle the flow and pressure requirements. The ruggedness and reliability of the compressor were established in two individual testing programs which, combined, lasted over 2 years, including 6,200 hours of operation with process gas mixtures. Maintenance problems were found to be minimal and mostly confined to the compressor ball bearings. Even when maintenance was required, the compressor, because of its compactness, was easily disconnected from the test site and moved to a rebuilding room.

COMPRESSOR DEVELOPMENT

The first approach to compressor type selection was to determine approximately how the proposed recycle compressor compared with conventional machines. Specific speeds and diameters were calculated from the head and suction volume requirements using a range of reasonable speeds and impeller diameters and assuming one, two, or three stages of compression. The speed range used in each case was 10,000 to 15,000 rpm, and the diameters were varied from 6 to 14 inches. When superimposed on a graph of

generalized compressor characteristics taken from Balje', all of the specific speeds (N_S) and specific diameters (D_S) fell in the high-head, low-flow range of the plot. This overlay of points for the hydrogen chloride recycle compressor is shown in figure 1. This plot should be used only for approximations, since Mach and Reynolds number effects and geometrical differences between compressors were neglected. In general, however, it can be seen that the hydrogen chloride compressor requirements fall in the region of the drag, or peripheral, compressor. This was expected, since the peripheral compressor is known to have high-head and low-flow characteristics. It can also be noted that the compressor efficiency in this region of operation is on the order of 20 to 50%. Although not shown, the fluorine recycle conditions lie in this region also.

Before proceeding further, a brief description of the peripheral compressor may be helpful to readers unfamiliar with this type machine. Typical features of the compressor are shown in figure 2, which is a photograph of an ORGDP experimental compressor. In the usual machine, the fluid enters at the periphery, passes through a stationary channel at the periphery of the impeller (gaining energy from the impeller), and discharges at the periphery. Between the suction and discharge is a stripper or block seal that minimizes leakage flow from the high pressure outlet region to the low pressure inlet region. The clearances between the impeller and the casing at the stripper and at the sides of the impeller are on the order of a few thousandths of an inch in order to keep the leakage losses as low as possible.

The actual pumping mechanism is not completely understood, which explains the several names given to the compressor: drag, regenerative, turbulent, friction, traction, etc. The results of experimental studies at the ORGDP†, however, support the regenerative theory, which states that the fluid enters the impeller blading near the inner radius or root and leaves at the tip, gaining momentum through centrifugal action. This process may be repeated several times between inlet and outlet, resulting in a general helical or corkscrew pattern of flow around the periphery. This multistaging effect is believed to be the reason for the high pressure ratios and low efficiency observed with the peripheral compressor.

Experimental data on the single-stage peripheral compressor, shown in figure 2, were available from previous work at the Oak Ridge Gaseous

* Balje', O. E., A Study on Design Criteria and Matching of Turbo-Machines; Part B. Compressor and Pump Performance and Matching of Turbocomponents, ASME Paper No. 60-WA231 (November, 1960).

† Cates, P. S., A Study of the Peripheral Compressor, Union Carbide Corporation, Nuclear Division, Oak Ridge Gaseous Diffusion Plant, August 26, 1966 (K-1665).

Figure 1
APPROXIMATE $N_s D_s$ -DIAGRAM FOR SINGLE-DISK PUMPS AND
LOW-PRESSURE-RATIO COMPRESSORS (Balje)

Figure 2
SINGLE STAGE PERIPHERAL COMPRESSOR

Diffusion Plant*. These data indicated the ability of the peripheral compressor to meet the compression requirements of both the hydrogen chloride and fluorine recycle applications. Based on this information, it was predicted that a two-stage machine having 10-inch-diameter impellers and operating at about 11,000 rpm with interstage cooling would be necessary for each application. The low flow required for the fluorine compressor caused some concern regarding low efficiency and, hence, high temperature rise. A concept to alleviate this problem (described later) seemed feasible, however, so that an acceptable compressor efficiency of 30% appeared attainable for both applications.

Although the gas-bearing approach for containment and rotor support looked interesting for these applications, it was decided to follow a path of better developed technologies. A design concept consisting of a shaft seal with externally mounted ball bearings and drive motor was therefore chosen. The inleakage requirements were not particularly stringent, but the reliability of the shaft seal was important. Also, it was necessary to establish the mechanical integrity of the high speed rotor, bearings, and motor.

The compressor development project, therefore, consisted of (a) evaluation of the compressor aerodynamics; (b) development of a shaft seal design; and (c) combination of the aerodynamic, seal, shaft, bearing, and motor designs into a reliable, complete machine. These items are discussed in detail in the following sections.

Compressor Aerodynamics

The aerodynamic design problem consisted of determining the optimum flow channel and impeller configuration for each application. The design philosophy was to meet the requirements of the worst cases in table I; i.e., lowest molecular weights and highest flows. Nozzle designs and impeller-to-casing clearances were essentially established from existing data. Two stages of compression with 10-inch-diameter impellers were tentatively selected for both mechanical and aerodynamic reasons.

Part of the ORGDP peripheral compressor program included studies of channel and impeller geometries. Since impeller geometry changes appeared to have potential for performance improvement, several designs were evaluated. Three conventional impellers, designated numbers 1, 2, and 3, with axial blade depths of 0.500, 0.375, and 0.250 inch, respectively, are shown in figure 3. Performance data for each of the three impellers measured on air at 11,000 rpm are presented in figure 4. Test results demonstrated that impellers with shallower blading provided cooler operating machines because of less carryover of high temperature discharge gas. For example, at 100 cfm, impellers 2 and 3 had 10 and 27% lower temperature rise,

* Cates, P. S., Peripheral Compressor Performance on Gases with Molecular Weights of 4 to 400, ASME Paper No. 64-WA/FE-25 (1964).

Figure 3
PERIPHERAL IMPELLERS

Figure 4
 PERIPHERAL COMPRESSOR IMPELLER CHARACTERISTICS

respectively, than did impeller 1. The shallower bladed impellers also produced lower pressure ratios, as can be seen in figure 4. For this reason, of the impellers tested, the one with the 0.500-inch-deep blading was judged best for this application.

Testing of several impellers with forward-bent blading at angles of 90, 75, 60, and 45 degrees (measured from the plane of the impeller) showed that the 60-degree bladed impeller was best. Consequently, a 60-degree bladed impeller with 0.500-inch-deep blading was incorporated into the prototype compressor design. Figure 5 shows three of the impellers tested.

Stationary flow channel dimensions were systematically varied and tested in air with the conventional 90-degree, 0.500-inch-deep bladed impeller. Figure 6 summarizes the results obtained in air at 11,000 rpm on nine channel designs which were closest to those preferred for the recycle compressor. Pressure ratio is plotted as a function of a dimensionless flow parameter which accounts for different fluid properties, defined as

$$\phi_t = \frac{W}{D^2 P_s} \sqrt{\frac{R_o T_s}{kg_c MW}}$$

where W = mass flow, lb_m/sec ,

D = impeller diameter, inch,

P_s = suction pressure, psia,

R_o = gas constant, 1,545 ft-lb_f/mole-°R,

T_s = suction temperature, °R,

k = ratio of specific heats,

g_c = gravitational constant, 32.17 lb_m-ft/lb_f-sq sec, and

MW = molecular weight.

Assuming two-stage operation, the approximate design points (air equivalent) for each stage of each compressor are included in figure 6. These channel data were obtained with a 0.015-inch impeller-to-casing clearance, which has proved to be both aerodynamically and mechanically satisfactory. The curves shown in figure 6 illustrate an important advantage of the peripheral compressor: With a basic impeller and housing design, the performance at a given speed can be varied greatly by the use of simple inserts to change the channel dimensions.

PHOTO NO. PH-65-633

Figure 5
EXPERIMENTAL IMPELLERS WITH FORWARD-BENT BLADING

Figure 6
 PERIPHERAL COMPRESSOR CHANNEL CHARACTERISTICS

It became apparent early in the program that two stages of compression would be necessary to permit reasonable operating speeds. Since the two sides of a peripheral impeller are independent, mounting two stages on a single wheel with some interstage sealing device seemed feasible. This dual stage, single impeller concept reduces the overhang moment on the shaft and improves the rotor dynamic characteristics. It also alleviates the problems associated with the low flow requirements of the fluorine application. The concept proved successful in performance tests.

Figure 7 shows the interstage labyrinth-type seal which was employed, along with a comparison of two impeller designs. The new impeller design illustrated there is only 60% as heavy as the original version and has calculated maximum stresses only 59% as great as the conventional solid-disc type impeller. Additionally, the interstage sealing length for the new design is twice as great, reducing interstage leaking; therefore, the new type impeller was included in the prototype compressor design.

Seal Development

As shown in table I, the maximum permissible leakage into the process is 0.2 scfm, with no outleakage tolerated. For the relatively high shaft speed necessary for these compressors, a positive clearance shaft seal was considered more feasible and reliable than a contact seal. Three seal types studies in the program were (a) labyrinth, (b) screw, and (c) bushing or smooth hub. Several designs of each type were evaluated on a test stand using a 1.9375-inch-diameter shaft size. Figure 8 is a photograph showing each seal type studied. All tests were made in air at speeds of 0, 8,600, or 11,000 rpm. Capacitance-type clearance probes were used to monitor shaft position under various operating conditions. Nominal radial clearances of 0.0020 and 0.0032 inch were tested; other geometric variables included length of seal, eccentricity of hub to sleeve, number and shape of labyrinth teeth, and design of screw seal grooving.

Testing of all three seal types showed the best for these applications to be the bushing seal. Table II summarizes the performance of the bushing seal. The flows reported are actual cubic feet per minute at 1 psi differential pressure. The effect of eccentricity was studied to predict the result of any misalignment upon assembly or operation. As may be seen in the table, even with a 3.2-mil radial clearance and a 0.5 eccentricity, a 2-inch-long seal is adequate. The shorter length and larger clearance increase the freedom from assembly and operating difficulties. With precision ball bearings, a 3-mil radial clearance was considered quite practical. Also, the pressure differential of 1 psi was considered adequate to prevent any backdiffusion of the process gas outward through the seal past the buffer gas (nitrogen).

Figure 7
TWO STAGE PERIPHERAL COMPRESSOR IMPELLERS

Figure 8
EXPERIMENTAL SHAFT SEALS

TABLE II

PERFORMANCE OF 1.9375-INCH-DIAMETER BUSHING SEAL

Design	Leakage Flow, cfm at 1 psi	
	2 Inches Long	3-1/2 Inches Long
1. Required*	0.155	0.155
2. 0.0020-Inch Radial Clearance (Concentric)	0.018	0.010
3. 0.0032-Inch Radial Clearance		
a. Concentric	0.096	0.054
b. Eccentric ($\epsilon = 0.5$)†	0.130	0.075

* Maximum permissible leakage based on 18 psia average process pressure.

† ϵ is eccentricity ratio which is zero when hub is concentric with sleeve and unity when hub is touching sleeve.

Developed Peripheral Compressor

The final compressor design, as derived from the development program, is shown in figures 9 and 10*. As noted on figure 10, for the sake of clarity, the gas nozzles are located slightly out of position. Through the use of removable inserts to change the channel dimensions, it appeared that one basic compressor design could meet the requirements for both hydrogen chloride and fluorine recycle service. Thus, the basic design consists of an impeller and motor rotor mounted on a single shaft supported in grease-packed precision ball bearings. A nitrogen-buffered bushing seal isolates the bearings and motor from the process gas with a small amount of nitrogen inleakage. Motor and bearing cooling is accomplished by a water jacket around the motor stator and an air thermal barrier between the compressor passages and the load bearing housing.

Prior to construction of the prototype compressor, shown in figure 9, a test rig was used to conform the performance of the integrated design in air. This rig was similar in appearance to the final prototype but was constructed of steel instead of nickel parts and was equipped with capacitance clearance probes to monitor shaft and impeller movement. The main

* Complete compressor details can be obtained in the Atomic Energy Commission CAPE Package No. 1624.

Figure 9
TWO-STAGE PERIPHERAL COMPRESSOR

Figure 10
PERIPHERAL COMPRESSOR

purpose of the rig was to optimize flow channel dimensions and to establish the mechanical integrity of the machine under various conditions of speed, loading, and cooling.

The final bearing design determined in the test rig consisted of a tandem mounted duplex pair of angular contact bearings for the load and a single angular contact bearing for the rear, with an axial spring preload of 100 pounds. The bearings were ABEC-9 (super-precision) and contained a channeled phenolic ball retainer for better grease retention (Fafnir* type 2MM9110W0-CRDU for the load and 2MM9106W0-CR for the rear). The bearings were lubricated with Andok 260† grease. Although the load bearing speed and size were near the usual upper limit for grease lubrication, the relative cleanliness and simplicity made the approach worthwhile.

The performance of the rig was excellent. Seal and impeller-to-casing clearances were easily set and maintained. Rotor dynamic characteristics were very good, with no critical speeds or excessive vibrations encountered. This was due primarily to precision machining and careful balancing of the rotors. Bearing temperatures were quite acceptable, with the load bearing running at about 145°F and the rear at 115°F under design conditions and at 11,000 rpm. Adequate motor and bearing cooling was achieved with 2 to 4 gpm of water to the motor and 30 to 60 cfm of air to the thermal barrier.

The channel and impeller designs tentatively selected from single-stage data were confirmed and optimized for the dual-stage/single-wheel configuration in the test rig. The final predicted performance curves for the hydrogen chloride and fluorine applications are shown in figures 11 and 12. These data were obtained at the design impeller tip Mach number, which is defined as

$$M_T = \frac{U_T}{C_s} = \frac{\pi D N / 720}{\sqrt{kg_c T_s R_o / MW}}$$

where U_T = impeller tip speed, ft/sec,

C_s = gas sonic velocity at suction conditions, ft/sec, and

N = rotational speed, rpm. (Other quantities were defined previously.)

For 11,000-rpm operation with 10-inch-diameter impellers, $M_T = 0.374$ for both the hydrogen chloride and fluorine applications. In air at room temperature, the appropriate test speed was 9,850 rpm. By slightly reducing the operating speed (easily accomplished with variable frequency

* The Fafnir Bearing Company, New Britain, Connecticut.

† Humble Oil and Refining Company, Houston, Texas.

Figure 11
 PERIPHERAL COMPRESSOR DUAL STAGE RESULTS -
 HCI DESIGN

Figure 12
 PERIPHERAL COMPRESSOR DUAL STAGE RESULTS -
 F_2 DESIGN

supply), the performance curve can be made to pass through the design point; furthermore, variable speed capability adds flexibility in meeting changing operating conditions.

Isentropic compressor efficiencies (defined as the ideal temperature rise in an isentropic process divided by the actual temperature rise) predicted for each stage at the design points were as follows:

	<u>Hydrogen Chloride</u>	<u>Fluorine</u>
First Stage	42%	34%
Second Stage	39%	34%

COMPRESSOR TESTING AND EVALUATION

Upon completion of the development program, the prototype peripheral compressor, illustrated previously in figures 9 and 10, was fabricated so that the integrated unit could be dynamically evaluated under design process conditions. The goals of the prototype testing project were to confirm the predicted performance characteristics of the compressor, to determine the reliability of the bearing and seal designs, and to establish maintenance and service requirements.

Because of its durability and strength in corrosive atmospheres, nickel was selected as the major material of construction for compressor parts exposed to the process gases. The impeller was machined from Duranickel. Extreme care was taken during assembly of the compressor to ensure that proper fit and clearances were maintained, since close tolerances were required*. Additionally, in view of the relatively high speeds involved, the rotor assembly was precision balanced.

Compressor Test Loop

To simulate process conditions, the test loop shown schematically in figure 13 was constructed using 2- and 3-inch-diameter Monel pipe. A charging manifold was provided through which air, nitrogen, and fluorine could be selectively introduced. A calibrated sharp-edged orifice was used to measure the flow rate of compressed gas, and two control valves, CV-1 and CV-2 in figure 13, were added to permit selection of the desired compression ratio across the compressor. Two sintered nickel filter tubes were placed in the loop upstream of the compressor to remove entrained solids. To compensate for the nitrogen seal feed inleakage to the test loop, a needle valve was installed on the high pressure side of the system to bleed off excess inventory. Correspondingly, a fluorine makeup system was added to allow fluorine concentration control.

* A complete assembling procedure is given in the appendix of this report.

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Figure 13
PERIPHERAL COMPRESSOR TEST LOOP

The peripheral compressor, as installed in the test loop, is shown in figure 14. Input services to the compressor included (a) 220/440 volt variable frequency alternating current, (b) 50 psig nitrogen, (c) 100 psig plant air, and (d) process cooling water. Since the compressor is driven by an alternating current induction motor, the speed of the compressor is approximately proportional to the frequency of the drive current. Therefore, a variable frequency power supply must be kept available if variable compressor speeds are desired. A 125/62.5 KVA, 100/50 kw, 80% power factor, 1,500/750 rpm, three-phase, 200/100 cycles, 440/220 volts, continuous duty, class B insulated generator was used to drive the compressor. The generator was directly coupled by means of an air-cooled eddy current coupling to a 200 hp, 220/440 volt, 460/230 amperes, 1,767 rpm, three-phase, 60-cycle, continuous duty, class B insulated, squirrel-cage induction motor.

The shaft seal flow instrumentation is shown schematically in figure 15. Dry nitrogen, supplied at 50 psig and reduced to 20 psig, was used to isolate the motor portion of the compressor from the corrosive process gases. The desired pressure drop across the seal and, hence, the distribution of seal feed gas, was set by the control valves on the feed and exhaust lines of the seal system. The seal pressures were controlled from the measurement of process pressure near the hub of the impeller.

As mentioned previously, an air flow was maintained in the region between the motor and compressor rotor housings to provide a thermal barrier that prevents excessively high load bearing temperatures. For this purpose, 30⁺ cfm of air was supplied. Most of the operational noise associated with the compressor came from this cooling air as it was released.

As shown in figure 13, heat exchangers were provided after each stage to remove the heat of compression. Normal operation resulted in a 200 to 250°F temperature rise across each stage, and consequently, from 8 to 10 gallons per minute of process water was supplied to each exchanger to lower the temperature of the discharged gas to around 90°F. In addition, approximately 4 gpm of cooling water was supplied to the shell of the motor housing to cool the compressor motor.

In addition to all the standard flow, pressure, and temperature instrumentation shown in figure 13, several alarm devices were incorporated into the design. Temperature alarm points were installed on the load bearing, the rear bearing, and the stage 1 discharge. These alarms were set to shut the compressor down in case of bearing or control failure. Since the normal operating temperatures of the load and rear bearings were expected to be 145 and 115°F, respectively, the alarm points were set at 190 and 150°F. The stage 1 discharge temperature alarm point was set at 450°F. An alarm bell actuated by low pressure was also connected to the suction side of the compressor to warn of possible malfunction of the pressure control valves or in the fluorine makeup system.

Figure 14
PERIPHERAL COMPRESSOR INSTALLED IN THE TEST LOOP

Figure 15
PERIPHERAL COMPRESSOR SHAFT SEAL FLOW INSTRUMENTATION

Compressor Testing Procedure and Results

The first testing program for the prototype recycle peripheral compressor was initiated with the objective of operating for a total of 2,000 hours on mixtures of nitrogen and fluorine at compressor speeds from 9,000 to 11,000 rpm while maintaining the required 2:1 compression ratio. Initial startup was at low speed (approximately 3,600 rpm) in order to prevent bearing grease churning and excessive bearing temperature rise (i.e., to "run in" the bearings). Because of an electrical instability in the power unit used to drive the compressor during the first testing program, compressor speeds were limited to around 10,000 rpm instead of the intended 11,000 rpm.

After completion of shakedown operations, mixtures of nitrogen and fluorine, with fluorine concentrations ranging up to 75%, were charged into the test loop and were recirculated. Normally, suction and discharge pressures were maintained near 12 to 14 psia and 28 to 29 psia, respectively. Gas flow rates ranged from 40 to 75 scfm, depending upon compressor speed, compression ratio, and molecular weight of the process gas. Table III, presenting a summary of pumping rates as a function of fluorine concentration, outlines the range in which the compressor was operated. All flows were measured with the compressor operating at 9,950 rpm* with a 2.1 pressure ratio. Testing periods for the compressor varied from only a few hours to over 100 hours of continuous operation. Because of the close radial tolerances of the interstage labyrinth seal, around 30 minutes was taken to start the compressor and to bring it up to speed and full load conditions so that uneven heating of the impeller and impeller housing could be prevented.

In order to compare the performance of the prototype compressor with that predicted from the test rig experiments, a series of runs was made at the design impeller tip Mach number (0.374). With nitrogen as the test gas, the approximate speed was 10,020 rpm for inlet temperatures of about 90°F. At these conditions, flow through the compressor was varied to give the data shown in figure 16. When compared with the predicted performance curve (taken from figure 12, page 33), it is seen that agreement was excellent.

Calibrated rotameters were installed in the nitrogen seal feed and seal exhaust lines to determine the seal flow distribution for various pressure differentials across the seal. In normal operation, pressure differences of 1.5 psi between the seal feed and the exhaust were maintained. Results of the flow determination, shown in table IV, demonstrate that the nitrogen seal inleakage to the process met the prescribed 0.2 scfm maximum.

* An alternating current drive frequency of 167 cps resulted in a compressor speed of 9,950 rpm.

TABLE III

VARIATION OF PROTOTYPE PERIPHERAL COMPRESSOR
PUMPING RATE WITH FLUORINE CONCENTRATION

<u>Fluorine Concentration, %</u>	<u>Molecular Weight</u>	<u>Pumping Rate*, scfm</u>
0.0	28.0	51.0
25.7	30.6	62.5
30.4	31.0	63.3
45.7	32.6	67.1
52.0	33.2	71.9
66.4	34.6	73.2
75.4	35.5	73.8

* Pumping rate for the compressor operating at 9,950 rpm with a 2.1 pressure ratio.

TABLE IV

SEAL FLOW DISTRIBUTION

<u>Pressure Difference, psi</u>		<u>Total Seal Flow, scfm</u>	<u>Exhaust Flow, scfm</u>	<u>Inleakage to Process, scfm</u>
<u>Between Seal Feed and Process</u>	<u>Between Seal Feed and Exhaust</u>			
1.00	0.50	0.09	0.05	0.04
1.50	0.50	0.15	0.05	0.10
1.50	0.20	0.09	0.00	0.09
2.00	0.50	0.15	0.04	0.11

Figure 16
COMPARISON OF ACTUAL AND
PREDICTED COMPRESSOR PERFORMANCE

After the compressor had been in service for a total of 321 hours, 60 hours of which involved recycling gas mixtures of nitrogen and fluorine, an unscheduled, sudden shutdown resulting from a malfunctioning temperature-recorder caused an unknown amount of fluorine gas to be forced past the seal system and into the bearing and motor area of the compressor. Until the time of the shutdown, the nitrogen feed to the seal was being supplied at a pressure of 5.0 psig. When the compressor tripped out of service and the test loop reached equilibrium pressure, the compressor housing was momentarily pressurized to 9 psig, causing outleakage through the seal. The nitrogen seal feed and exhaust rotameters located some 10 feet away were etched by the action of the fluorine, thereby establishing the fact that fluorine gas had reached the motor area and hence passed over the load bearings of the compressor.

Following the exposure, the nitrogen seal supply pressure was raised to 20 psig, and the compressor was restarted without incident. For the next several hundred hours, the bearing temperature continued to be normal, denying any evidence of fluorine attack. After 320 hours of operation on fluorine (over 2 months later), however, the compressor load bearing began to overheat in a sporadic cyclic fashion, going from a normal temperature of 138°F to as high as 180°F. The cyclic overheating varied in duration from 20 minutes to over 1 hour, and the elapsed time between the end of one cycle and the beginning of another was as short as a few minutes in some instances and almost 8 hours in other cases. The pumping performance of the compressor and the temperature of the rear bearing appeared unaffected. After 27 thermal cycles, a syringe was connected to a lubricating tap located on the underside of the compressor, and 7 cc of oil was injected into the area of the load bearing. The bearing responded immediately to the auxiliary lubricant and, within several hours, was operating normally.

The peripheral compressor continued to run without trouble for 670 hours after the first oil injection. Then, as before, the load bearing began to overheat, and consequently, 2 cc of oil was injected into the compressor at the load bearing tap. The bearing again responded, and compressor operation was continued toward the 2,000-hour goal.

Later, after an additional 443 hours of operation on fluorine (1,433 hours total operation on fluorine), the load bearing went through three short thermal cycles and leveled out at a steady 145°F, 15°F higher than the previous normal temperature of 130°F. Two cubic centimeters of oil was again injected into the bearing area in an attempt to reduce the bearing temperature to around 135°F. This time, the oil injections had little effect, and the bearing temperature dropped only slightly to 143°F, where it remained to within 1°F. Thermal cycling was not apparent, however, so operation was continued. At this point, another abnormality was noticed; i.e., the rapid manner in which the load bearings heated up to running temperature during startup operations. The bearing temperature rose at a rate which was two to three times greater than observed previously.

Eight months after initiation of the test, with a total of 2,206 hours of operation, including 1,809 hours on fluorine, the alarm circuit monitoring the load bearing temperature (set to respond at 190°F) tripped the compressor out of service. A wattmeter recording the power input to the compressor indicated that the bearings were seizing or grabbing prior to the shutdown. The rear plate was removed from the compressor, and the drive shaft was rotated by hand. The shaft rolled freely, without roughness or undue resistance to rotation. Two cubic centimeters of lubricating oil was again injected into the bearing set, and the compressor was started. Startup was smooth and without any indication of bearing damage other than the rapid rate at which the load bearing attained its operating temperature.

The testing program was terminated after a total of 2,413 hours of operation, including 2,016 hours of corrosive gas service. The peripheral compressor was removed from the test loop and was disassembled. Careful inspection was made of every component as it was removed from the compressor, with particular attention focused on evidences of fluorine attack.

The front plate covering the impeller housing was removed first. The inside of the housing was very clean, and all metal surfaces exposed to the fluorine gas still retained their luster and appeared to be in the as-installed condition. The front plate covering after disassembly is shown in figure 17. The labyrinth interstage seal and Duranickel impeller were then removed and inspected. Pictures of the impeller and impeller housing, figures 18 and 19, are evidence of the immaculate condition of the surfaces exposed to the process gas. The minute scratches and drill holes seen on the Duranickel impeller resulted from machining and balancing prior to assembly.

The rear plate was removed from the back side of the compressor motor housing to permit access to the outer bearing. Inspection of the bearing in place showed the grease to be stiff but not caked on the outside, but powdery around the bearing balls.

The motor housing was separated from the impeller housing subassembly, and then the motor shaft holding the rear and load bearings was pulled from the impeller housing. Inspection of the motor housing containing the stator showed that all surfaces forward of the windings, as well as the front portion of the windings themselves, were coated with a very black and gritty looking oil. The motor was electrically tested and was found to be in good operating condition.

The load and rear bearings were pulled from the motor shaft and were partially disassembled, as shown in figures 20 and 21. Both load bearing retainers were covered with a thin layer of a black, gritty, grease-like material. The inside bores of the two load bearing races were not discolored, indicating from previous experience that the load bearings had not overheated too severely during the testing program. Only a minimal amount of grease was found packed behind and around the balls of the load bearings, but the rear bearing was found to be only slightly deficient

Figure 17
COVER PLATE ASSEMBLY AFTER TESTING

PHOTO NO. PH-66-1791

Figure 18
IMPELLER AFTER TESTING WITH FLUORINE

Figure 19
IMPELLER HOUSING AFTER TESTING WITH FLUORINE

PHOTO NO. PH-66-1787

Figure 20
LOAD BEARINGS AFTER TEST

Figure 21
REAR BEARING AFTER TEST

in the quantity of grease present. Grease samples taken from the load and rear bearings and submitted for analyses yielded fluoride contents of 6.1 and 2.0%, respectively.

After cleaning, a detailed inspection was made of the bearing sets. Final inspection found the rear bearing to be in excellent condition. The load bearings, however, bore a number of marks indicative of their hard service during the year. The back load bearing (nearer the motor) appeared to have been subjected to more heat or less lubrication than the forward bearing (nearer the impeller), as the surfaces of the back ball retainer were rough and darkened*, and the balls were slightly discolored. The inside circumference edge of the forward ball retainer was found burned, or charred, and was cracked in three places. The burned area extended almost halfway down the inside and almost halfway across the radial thickness of the retainer, as shown in figure 22. The reverse side of the ring showed no signs of attack. The burned edge of the forward retainer was nearest to the process and consequently was the first to be exposed to the corrosive process gases in the event of a seal failure. Inspection of the balls from the forward load bearing revealed that all ball surfaces were moderately scratched. Although discolored, the back ball set still displayed a polished surface. Apparently, the charred phenolic bearing retainer ring supplied the abrasive material responsible for the scratching action sustained by the forward ball set.

For comparison, the bearings used in the test rig, described in a previous section, were removed and inspected. To provide more information on the performance of the bearing design, this rig had been operating continuously in air at 11,000 rpm for 2,215 hours under load conditions simulating those of the prototype recycle compressor. These bearings were in very good condition, with the grease and ball separators retaining their original color. Although the grease had lost some of its oil, it still had a good degree of lubricity. No makeup oil had been supplied to the bearings.

After inspection, the prototype peripheral compressor was reassembled with new bearings and was once more installed in the test loop. A modification was made to the test loop so that low concentrations of gaseous uranium hexafluoride could be maintained in the system. For the second testing program, no target was set for the number of hours of operation. The compressor was to be operated in the same general way as before but, if possible, without an exposure of the motor section to corrosive gases, as experienced inadvertently during the early stages of the first test.

The initial startup with new bearings was again performed slowly to prevent grease churning and excessive bearing temperature rise, as mentioned previously. Approximately 30 hours of compressor operation under reduced load conditions was logged on the new bearings before the compressor speed and output requirements were raised to normal operating levels. Even then, bearing temperatures were closely watched until normal operating temperatures were achieved.

* The ball retainer ring material is of the phenolic type.

Figure 22
BALL BEARING RETAINER RING AFTER TEST

The compressor performed well throughout the second test period. Mixtures of gas containing up to 3.0% uranium hexafluoride were charged into the test loop and were recycled. Generally, suction pressures of 13.0 psia and discharge pressures near 28.0 psia were maintained. The stage 1 and stage 2 suction temperatures normally were near 90°F, with stage discharge temperatures between 275 and 350°F, depending upon the density of the gas being pumped. For pure air or nitrogen, the discharge temperatures were 330 to 350°F, while for 3.0% uranium hexafluoride in nitrogen, the temperatures ranged from 275 to 300°F. No trouble was encountered with the seal or other control instrumentation during the test period.

After 3,813 hours of operation, including 2,340 hours of service with mixtures of gas containing uranium hexafluoride, the peripheral compressor was shut down because of an apparent load bearing failure. The compressor had functioned extremely well without any indication of failure until 2 days before the shutdown. As opposed to the first test series, the compressor had required no maintenance from the time the unit was put into service with its second set of bearings until 14 months, or 3,790 operating hours, later.

After approximately 3,760 hours of operational service, the load bearing temperature slowly increased from 130 to 145°F. About 24 hours later, the bearing temperature, within minutes, rose to 160°F and then slowly began increasing. At 168°F, 1 cc of oil was injected into the bearing area. The bearing responded immediately with a 5°F temperature increase and, 1 hour later, was back to 168°F; however, the bearing did not respond further to the auxiliary lubricant and did not cool to below the relatively hot 168°F level. At this point, the compressor was shut down for fear of possible damage if the load bearing suddenly seized, since it was evident that one of the tandem-mounted load bearings was damaged.

The peripheral compressor was again removed from the test loop and was disassembled. Unusual difficulty was encountered in tearing down the process side of the compressor, with the impeller being especially difficult to remove. All surfaces contacting the process gases were covered with an adherent layer of uranyl fluoride. Such deposits, varying in color from light yellow to green, are typical of equipment used in uranium hexafluoride service. Dusting, however, was not apparent. The impeller and impeller housing are shown, as removed, in figures 23 and 24. All metal surfaces contacting the corrosive gases were easily cleaned with water, and close inspection showed no signs of corrosive attack.

The general condition of the rear bearing can be described as excellent. The grease packed throughout the bearing, although slightly stiff, appeared to possess much of its initial lubricating quality. The balls, phenolic retainer ring, and bearing races were in good condition.

Inspection of the two load bearings showed that the load bearing nearer the compressor motor had failed. The evidence classified the bearing failure as a normal one, resulting from lack of lubrication due to a breakdown of the bearing grease through normal use. Although the forward bearing was deficient in the amount of grease packed around the balls,

Figure 23
IMPELLER AFTER TESTING WITH URANIUM HEXAFLUORIDE AND FLUORINE

PHOTO NO. PH-68-543

Figure 24

IMPELLER HOUSING AFTER TESTING WITH URANIUM HEXAFLUORIDE AND FLUORINE

the bearing was not thought to be dry and did not appear damaged. The phenolic ball retainer ring of the motor side load bearing was found broken in one place. The surfaces of the inner and outer races on which the balls ride were found to be moderately spalled. The damaged surface of the inner race is displayed in figure 25. All balls of this bearing were slightly marred, with the surface of one ball slightly chipped. As noted previously, an oil injection was made after trouble was ascertained; by then, however, the bearing damage most likely had already occurred. Possibly, the life of the bearing could have been increased by an earlier injection of auxiliary lubricant, say after 2,000 hours of operation.

The peripheral compressor was reassembled with new bearings and was placed back into the test loop. No difficulty was encountered in putting the compressor back together. The unit was started and operated for several hours under a no-load condition with the new bearings and was then put back into full service. Currently, the compressor is being used for another project requiring corrosive gas recycle.

CONCLUSIONS AND RECOMMENDATIONS

The prototype peripheral compressor, as evaluated from over 2 years of intensive testing, including over 6,200 hours of actual operation, has proved worthy of high recognition as a potential recycle compressor for the proposed volatility process. The performance of the prototype compressor exceeded the design criteria set forth for the fluorination step of the process. Although the machine was not actually tested under the conditions specified for the hydrochlorination step of the volatility process, there is no doubt that the prototype compressor could adequately handle the flow and pressure requirements. The ruggedness of the compressor was demonstrated during the first testing program where the load bearing and motor area of the compressor were inadvertently exposed to fluorine, and the compressor continued to perform for over 8 months with periodic oil injections. The reliability of the machine was established during the second testing program in which the compressor was operated for 14 months without need for maintenance. The compressor, being relatively small and compact, could easily be disconnected and removed from the process site for bearing replacement or other desired maintenance service. In the event of a seal failure, contaminated or corrosive gases could not be expelled to the atmosphere because the compressor passages and the drive motor comprise an integrally sealed package.

The peripheral compressor, however, is not without fault. First to be considered is the fact that the compressor requires close machining tolerances, and like any other precision piece of equipment, a certain amount of care must be exercised in the construction and operation of the unit. Precision machining and balancing are mandatory, and special care must be employed in the assembly of the compressor to ensure that proper movement and tolerances are respected. Consideration must be given to the possibility that the impeller can contact the interstage seal during rapid

PHOTO NO. PH-68-622

Figure 25
LOAD BEARING INNER RACE AFTER TEST

startup due to uneven heating. Also, a filter similar to the sintered metal one used in the test loop is highly advisable upstream of the compressor to remove entrained solids in the pumping media.

The impeller and motor rotor are mounted on a single shaft supported in grease-packed precision ball bearings. The bearings must be replaced periodically, most likely around every 2,500 hours if operated intermittently at 10,000 to 11,000 rpm. Bearing life would be expected to be longer under continuous operation and especially at lower speeds. Also, periodic injection of makeup oil (say, every 1,000 hours) should significantly extend bearing life. With experienced mechanics, the compressor can be removed from the process site, disassembled, equipped with new bearings, and placed back into service within 1 or 2 days.

The prototype peripheral compressor is equipped with a bushing type shaft seal, and therefore, a certain amount of nitrogen inleakage to the process must be tolerated. If the process system is a closed one, as was the test loop, provisions must be made to bleed off excess inventory and to maintain desired reactant concentration levels by makeup to prevent dilution.

Lastly, since the peripheral compressor is driven by an alternating current induction motor, a variable frequency alternating current power supply is required to achieve selection of compressor speed. If variable speed is not desired, a source of 185 cps power is needed for 11,000 rpm operation.

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APPENDIX

PROTOTYPE PERIPHERAL COMPRESSOR ASSEMBLY PROCEDURE

The following assembly procedure is recommended for the peripheral compressor described in this report*. It is assumed that the motor rotor-to-shaft assembly and the rotor assembly balancing have been completed prior to these steps.

1. Rotor and Bearing Assembly

- a. Clean parts thoroughly and apply a very light oil coating to steel parts to prevent rusting.
- b. Pack bearings with Andok 260 grease, taking special effort to keep bearings clean. Pack each load bearing with about 4 grams and the rear bearing with 2 grams. Rotate bearings as they are packed, ensuring grease penetration to balls, races, and channel in ball retainer.
- c. Heat bearings to about 200°F to expand for easy assembly to shaft. (A heat lamp proved satisfactory.)
- d. Slip thrust plate onto shaft.
- e. Coat inner bearing races and shaft bearing surface lightly with an oil that is compatible with the grease used. With rotor vertical (large end up) and thrust plate supported to slightly below the level of the bearing shaft shoulder, slip on first load bearing with low shoulder on inner race toward motor. (This step is important to prevent bearing from separating.)
- f. Install load bearing spacers. Add about 1 or 2 grams of grease to cavity between spacers.
- g. Slip on second load bearing with low shoulder toward motor.
- h. Install outer spacer and grease seal.
- i. Screw on modified lock nut and tighten against bearing assembly. Be careful to prevent foreign matter from falling into bearing.
- j. Relocate rotor with small end up. Install spring assembly (previously set to give 100 pounds preload) and support so that top surface is slightly below bearing shaft shoulder.

* Reference drawing details in AEC CAPE Package No. 1624.

- k. Wipe light film of oil on bearing inner race and shaft surface. Slip on rear bearing with low shoulder toward motor. Screw on locknut and tighten, taking care not to get dirt in bearing.

2. Rotor and Housing Assembly

- a. Clean parts thoroughly; main compressor housing, motor housing, rear bearing housing, and end cap.
- b. Support main compressor housing with centerline vertical and motor side down about 3 feet off the floor.
- c. Wipe light film of oil in bearing housing and place 1 or 2 grams of grease in thrust plate recess. Install appropriate shim spacers determined from previous measurements for desired impeller-to-casing clearances. (The motor side impeller clearance should be 0.012 to 0.015 inch.) Supporting rotor by thrust plate, push rotor assembly vertically upward into housing. Dowel and tighten screws.
- d. Install O-ring in motor compartment seal, using a silicone grease or equivalent. With motor housing supported a few inches off the floor (with center open), lower compressor housing and rotor assembly into motor housing, dowel, and bolt.
- e. Support compressor and motor housing assembly about 3 feet off the floor with the motor end down. Install O-ring in rear bearing housing plate and wipe a light film of oil on bearing housing bore and bearing. Bring bearing housing plate up to point of assembly. With long No. 10-24 screws, grasp spring assembly and pull down while bearing housing is being pushed into place. Bolt bearing housing to motor housing. Replace long No. 10-24 screws with shorter screws and secure spring assembly to bearing housing. Be sure that spring assembly housing is in solid contact with bearing housing plate.
- f. Protect exposed bearing from dirt until later installation of end cap.

3. Impeller, Inserts, and Labyrinth Assembly

- a. Clean parts thoroughly and position compressor with centerline horizontal. Check radial clearance between shaft and seal housing; should be 0.0025 to 0.003 inch at all points. By appropriate measurements, check rear impeller-to-casing clearance (should be 0.012 to 0.015 inch, cold).
- b. Install first-stage stripper block and desired first-stage channel inserts using nickel-plated screws. (Shim behind stripper block with nickel shim as necessary to give 0.012 to 0.015 inch clearance with impeller blading.)

- c. Install impeller with dowels and nickel-plated steel or Duranickel bolt and washer.
- d. Install labyrinth, dowel, and position with nickel shims between outside diameter and casing to achieve 0.005-inch radial clearance between it and impeller at all positions.
- e. Install filler, second-stage inserts, and stripper block onto cover plate using nickel-plated screws. Install aluminum gaskets into cover plate grooves.
- f. Check depth from casing face to side of impeller and compare with height of filler above cover plate. The former measurement should be 0.015 to 0.018 inch greater than the latter. Make similar measurements in the stripper region. Shim as necessary with nickel.
- g. Assemble cover plate assembly with stripper on right side (looking toward impeller).
- h. Install O-ring in rear end cap and assemble end cap to rear bearing housing. Before tightening screws, install bearing thermocouple probe. After tightening, install speed pickup, setting pickup about 1/32 inch from shaft protrusions.

4. Startup

- a. Run in new bearings at about 3,600 rpm for at least 2 hours prior to high speed operation. Observe bearing temperatures for evidence of overheating. It is recommended that high speed be approached gradually, in steps of 500 to 1,000 rpm.
- b. Loading of compressor should be gradual to allow uniform heating of impeller and housing. About 30 minutes from wide open flow conditions to the 2-to-1 pressure ratio has been sufficient. Rapid loading may result in rubbing between the impeller and interstage labyrinth seal.