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Moore, R. W.; Shaffer, R. W.; and McCullough, J. E., "A-Scroll Compressor for Shipboard Helium Liquefier Systems" (1976).
International Compressor Engineering Conference. Paper 223.
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A SCROLL COMPRESSOR FOR SHIPBOARD HELIUM LIQUEFIER SYSTEMS

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I. INTRODUCTION

As part of its program in developing superconducting electrical machinery for ship propulsion, the U. S. Navy has established requirements for helium compressors to be used with shipboard helium liquefaction systems. For several years, Arthur D. Little, Inc. (ADL) has been working on the development of a proprietary new type of rotary-positive-displacement compressor, called the scroll compressor. In 1973, under an Office of Naval Research Contract, we investigated the applicability of scroll compressors to operation with helium for shipboard use, and evolved a design for an experimental 90 CFM, 16 to 1 pressure ratio, two-stage, oil-lubricated unit. Since 1974, under a contract from Naval Sea Systems Command (NAVSEA), we have been developing such a unit, and it has recently been delivered to the Navy for laboratory testing. This paper describes the results to date of our programs for the Navy.

II. DESCRIPTION OF SCROLL MACHINE

The basic elements in a scroll compressor are two identical spiral scrolls, each fixed to, or integral with a cover plate. When the two scroll-plate assemblies are mated, the two scrolls are interleaved in an eccentric fashion, so that they touch and form a series of crescent-shaped pockets between the two cover plates as shown in Figure 1. One of the scroll plate assemblies is fixed. The other one orbits around the center point of the fixed assembly; and as it moves, the pockets formed by the interleaved scrolls follow the spiral toward the center and diminish in size. The moving scroll plate assembly orbits with a fixed angular orientation; it does not rotate. It is driven by a simple short-throw crank mechanism.

Figure 2 shows the progress of the compression process. The compressor's inlet is at the periphery of the scrolls. The entering gas is trapped in a pocket and compressed as the pocket moves toward the center. The compressed gas is exhausted through the outlet at the center of the fixed spiral.

If the inlet and outlet ports, and the direction of orbital motion are reversed, the device can operate as an expander.

The scroll compressor appears to have several advantages for the Navy's application. The area-to-volume ratio of gas pockets is much higher than in a piston-cylinder arrangement. Therefore, cooling the scrolls, so as to maintain moderate wall temperature, can effectively remove heat from the gas during compression; oil-flooding the compression pockets is not necessary. The helium discharge temperature, which tends to be high because of the high ratio of specific heats, can thus be moderated and good compression efficiency achieved.

Because of the absence of valves, a major item in compressor maintenance is eliminated. The compressor provides a relatively continuous discharge flow and its rotating/orbiting parts can be effectively balanced; hence, low noise/vibration levels are possible. Scroll plates tend to wear in with operation, allowing long life. The machinery is relatively simple and rugged, with few moving parts, using common materials of construction, so that good reliability at acceptable cost should be achievable. It can readily be designed to meet the high shock loading requirements for shipboard equipment, and in a compact configuration.

The Navy requires less than 1 ppm impurity in the helium discharge of compressors for liquefier systems. It is normally met by machines with oil-lubricated compression spaces by incorporating oil removal equipment for the discharge gas as part of the compressor system.

III. EXPERIMENTAL SCROLL COMPRESSOR

The experimental, two-stage scroll compressor as shown in Figure 3 is laid out such that each stage is driven from one end of a common drive motor. This arrangement facilitates developmental testing and modifications as required during the development process. Each stage has its own bearing-mounted crankshaft which is connected with a

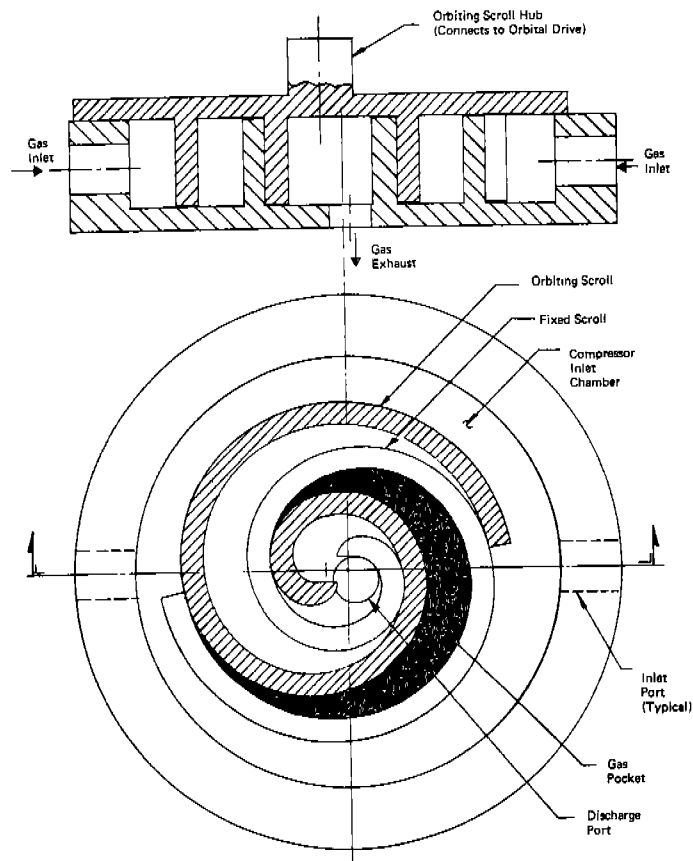


Figure 1

TYPICAL SCROLL COMPRESSOR CONFIGURATION--ORBITING SCROLL AT GAS POCKET SEAL-OFF POSITION

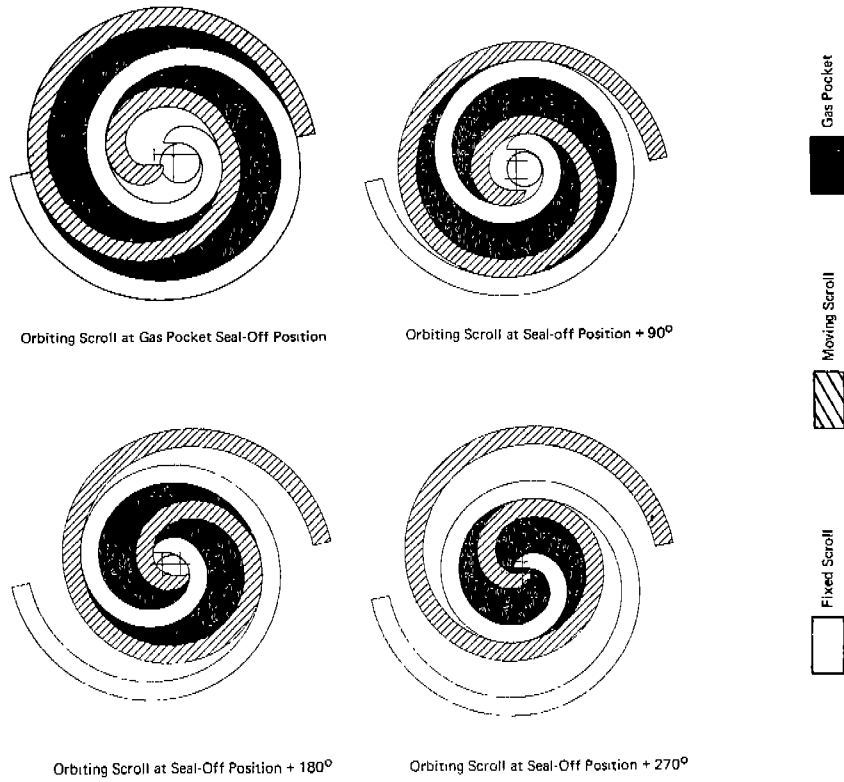


Figure 2

FOUR OPERATING POSITIONS OF ADL SCROLL COMPRESSOR/EXPANDER

flexible coupling to the double-ended shaft of the motor. Aftercoolers and oil traps are provided for both stages of the compressor and these items and piping are contained in the space within the frame below the motor and the two stages.

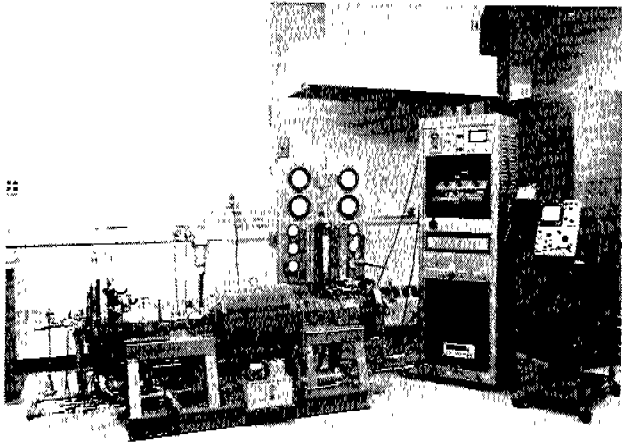


Figure 3

PHOTO OF ASSEMBLED COMPRESSOR

The two stages of the compressor are similar in design, the second stage being smaller due to the lower volume flow rate. Figure 4 shows the scrolls for the second stage, the orbiting scroll being on the left. A description of the first stage will serve for both. The fixed scroll is rigidly mounted to the outboard portion of the compressor housing. The interior of the housing is pressurized to provide a pneumatic load on the back of the orbiting scroll, sufficient to hold the two scrolls in close proximity. A sealing arrangement with axial compliance maintains sealing at the tips of the scrolls, despite variations in axial height of the involutes resulting from manufacture or radial temperature gradients during operation. A similar compliant seal arrangement is provided around the periphery of the fixed scroll to provide a seal between the pressurized housing and an inlet manifold for the stage.

The housing pressure is maintained at a level between inlet and discharge pressure, i.e., about 12 psig in the first stage and 100 psig in the second stage. A rotary shaft seal is provided where the drive shaft passes through the housing. Two process gas inlet ports near the outer perimeter of the fixed scroll and 180° apart connect to the inlet manifold as shown schematically in Figure 1. An Oldham coupling that prevents rotation of the orbiting scroll fits between the back of the orbiting scroll and the housing with a small amount of axial clearance to insure that it does not carry any axial load. An oil-lubricated thrust bearing is also provided between the back of the orbiting scroll and the housing. This bearing has two

functions: first, it takes some portion of the load from the orbiting scroll during transient conditions when the housing pressure might be low (such as during start-up); and second, stabilizes the orbital motion of the orbiting scroll.

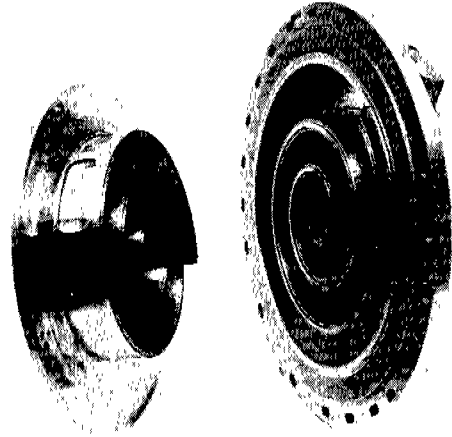


Figure 4

PHOTO OF SECOND STAGE SCROLLS

The fixed scroll is water cooled, utilizing passages within the scroll element. Thermal contact between the orbiting scroll and the fixed scroll and the oil lubrication on the back of the orbiting scroll cool this component.

The bearings and rotary shaft seals for the stages are oil lubricated by a positive feed system consisting of a standard lubrication metering pump.

Conventional materials of construction are used in each stage of the compressor. The housing is gray cast iron. The fixed scroll and its water jacket, nodular cast iron, the orbiting scroll, medium carbon steel (AISI 4140) and the shaft and parts for the drive mechanism low carbon steel. The involute tip seal element in each scroll is made of the same material as the scroll in which it fits. The materials for the fixed and orbiting scrolls were chosen in part, because of their excellent friction and wear characteristics in sliding contact with boundary lubrication such as occurs in this oil-lubricated machine. The thrust bearing behind the orbiting scroll is also made of nodular cast iron to achieve good friction and wear characteristics. The Oldham coupling is made from aluminum to minimize its weight and dynamic forces associated with its motion.

The electric drive for the two stages is provided by an industrial three-phase, 60 Hz, 220 volt, 1,800 rpm (synchronous speed) induction motor rated for 30 hp (Frame 286T).

IV. TEST SETUP

The test setup, shown in Figure 3, has the first stage on the left side of the mounting frame, the drive motor in the center, and the second stage on the right. Pressure gauges and flow meters are mounted on the panel to the right of the frame; electrical and electronic instruments and controls are in the cabinet to the right. The piping, valving and instrumentation permits either stage to be operated separately, and its performance measured, or both stages to be operated together. Helium process gas flow rates are measured by a Fischer & Porter rotameter. Process gas pressure and temperature instrumentation are located at the inlet and discharge of both stages and at the rotameter. A Dwyer rotameter measures the cooling water flow rate to each stage.

The drive motor is mounted by its double-ended shaft on pillow blocks at both ends; hence, the casing of the motor is free to rotate. It is restrained by a level arm connected to a load cell which provides a measurement of the reaction torque on the motor (equal to the drive torque for the compressor). Motor speed is indicated either by a stroboscope or by measuring the output frequency of an optical sensor on the shaft. The combination of torque and shaft speed is used to determine compressor input power.

Provisions were also made for obtaining p-V diagrams of the compression process. Pressure taps at selected points of the fixed scroll involute can be connected to a high-frequency-response pressure-transducer whose output is displayed on an oscilloscope. The sweep on the scope is triggered by the optical sensor on the motor shaft. Each pressure tap is exposed to the pressure variations within a pocket formed by the involute scrolls during 360° of rotation. With the scroll configurations on this unit, two pressure taps, one wrap apart along the involute surface, can indicate the entire compression process in a particular pocket. Two such sets of taps have been provided on each side of the fixed scroll, 180° apart, so the compression processes in the two sets of pockets passing through the machine, 180° apart, can be monitored. The p-V diagrams can be used to determine gas compression power; subtraction of compression power from shaft input power gives mechanical losses.

The performance achieved with the compressor operating on helium is summarized in Tables 1 and 2. Table 1 is for the first and second stage operating individually at their design point. Table 2 is for both stages operating together. Table 1 indicates the first stage has a higher volumetric efficiency than the second stage. With design inlet pressures on the two stages (i.e., 1-2 psig and 45 psig, respectively), the flow-rate through the first stage would be larger than that through the second. Thus, operating the two stages in series, with the same throughflow, would produce a higher intermediate pressure (or second stage inlet pressure) than desired. To avoid excessive intermediate pressure during two-stage operation, a small flow rate (estimated to be about 9 SCFM) was bypassed from intermediate pressure back to the first stage

suction. This bypass flow did not pass through the flow meter; hence, no credit is taken for it in the overall compressor performance shown in Table 2. If the first stage flow rate were perfectly matched to the second stage, as it should be in a future unit, we would expect the power input to the unit to be reduced by about 1.3 hp. The corresponding isentropic efficiency for the unit would then be 0.83.

V. FUTURE DEVELOPMENT

Phase II of our work for NAVSEA has recently begun, focusing on developing an advanced prototype compressor system including equipment for oil removal from the compressed helium. The two compressor stages will be similar to the Phase I designs, but a more compact overall arrangement will be achieved by coupling the orbiting scroll of both stages through swing-link mechanisms directly to opposite ends of the motor shaft. The separate drive shafts and associated bearings required for each stage of the Phase I unit will thus be eliminated. Other design improvements will include:

- An enclosed motor so that the compressor is hermetic with no shaft seals to ambient atmosphere.
- Better scroll cooling
- Two-piece scroll construction to simplify fabrication and provide internal damping in the scroll assemblies.
- Use of an oil-lubricated hydrostatic thrust bearing to take axial loading on the orbiting scroll--this eliminates the need for pressurized drive housings.

Oil cleanup equipment patterned after that now in use in laboratory-type, oil-flooded reciprocating compressors for helium will be used, including a bulk-oil separator, an oil-mist separator and a charcoal absorber, in that order.

An isometric view of the advanced prototype compressor system is shown in Figure 5. Salient characteristics are summarized in Table 3. The efficiency in Table 3 (and resultant power input) is thought to be conservative since stage efficiencies of about 0.8 have already been achieved in the Phase I compressor. A better cooling arrangement combined with elimination of some bearings and seals in the Phase II compressor should result in improved stage and overall efficiencies.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the contributions to this work by: Niels Young, the inventor of the scroll compressor; the ADL Invention Management Group, who supported early development activities; NAVSEA, for financial support of this program and the Naval Ship Research and Development Center (NSRDC), Annapolis, for technical direction.

Table 1

PERFORMANCE OF FIRST AND SECOND STAGE ON HELIUM

	<u>First Stage</u>	<u>Second Stage</u>
Inlet Pressure (psig)	1.5	
Discharge Pressure (psig)	45	45
Pressure Ratio	3.70	225
Gas Inlet Temperature (°F)	85	4.01
Gas Discharge Temperature (°F)	329	79
Cooling Water Temperature (°F)	41	333
Cooling Water Flow Rate (gpm)	2.6	41
Housing Pressure (psig)	12	2.5
Speed	1,790	100
Measured Flow Rate (SCFM)	98.14	1,790
Displacement Flow Rate (SCFM) (1)	101.47	89.80
Volumetric Efficiency	.967	94.00
Shaft Power Input (hp)	14.43	.956
Isentropic Power Input (hp)	11.20	13.45
Overall Isentropic Efficiency (2)	.776	10.80
		.803

(1) Based on volume of inlet pockets at seal-off.

(2) Based on shaft power input.

Table 2

TWO-STAGE PERFORMANCE ON HELIUM

First Stage Inlet Pressure (psig)	1.0	1.0	1.0
Second Stage Inlet Pressure (psig)	46	45	45.50
Discharge Pressure (psig)	249	226	196
Overall Pressure Ratio	16.82	15.35	13.44
First Stage Inlet Temperature (°F)	94	92	87
First Stage Discharge Temperature (°F)	333	333	332
Second Stage Inlet Temperature (°F)	96	96	97
Second Stage Discharge Temperature (°F)	362	344	313
Cooling Water Flow Rate (gpm)	5.0	5.0	5.0
Cooling Water Temperature (°F)	41	41	41
Speed (rpm)	1,770	1,770	1,770
Discharge Flow Rate (SCFM) (1)	86.50	86.10	86.90
Shaft Power Input (hp)	28.46	26.48	25.32
Isentropic Power Input (2)	22.02	20.90	19.59
Overall Isentropic Efficiency	.774	.789	.774

(1) Based on rpm, inlet pressure and temperature to second stage and volumetric efficiency of second stage determined from previous tests. Some flow from first stage bypassed second stage in this test to achieve proper second stage inlet pressure.

(2) Assuming equal pressure ratio per stage.

Table 3

ADVANCED PROTOTYPE 90 CFM SCROLL
COMPRESSOR SYSTEM CHARACTERISTICS

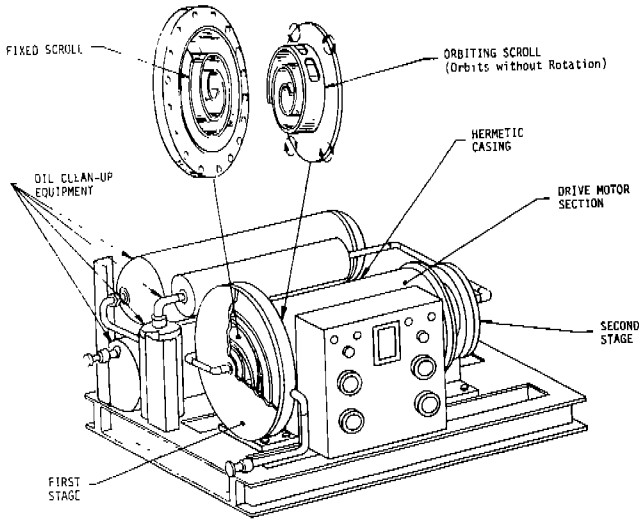


Figure 5

ADVANCED PROTOTYPE
90 CFM ADL SCROLL HELIUM COMPRESSOR

Flow Rate (SCFM)	90
Inlet Pressure (atm)	1
Inlet Temperature (°F)	100
Discharge Pressure (atm)	16
Compressor Adiabatic Efficiency	0.8
Shaft Power Input (hp)	28.0
Drive Motor Efficiency	0.9
Electric Power Input (kw)	23.2
Cooling Water Flow Rate (gpm)	5

Envelope Dimensions (with Oil-Cleanup Equipment)

Height (ft)	2.3
Width (ft)	3.5
Length (ft)	3.5
Volume (cu. ft)	28

Approximate Weight (lbs) 1,200