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**MECHANICAL REFRIGERATOR FOR DEEP WELL
SONDE ELECTRONICS, PHASE II**

Final Report

**By
J. Fukuzawa**

April 9, 1979

Work Performed Under Contract No. EY-76-C-03-1309

**System Development Corporation
Santa Monica, California**

**U. S. DEPARTMENT OF ENERGY
Geothermal Energy**

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MECHANICAL REFRIGERATOR FOR DEEP WELL SONDE ELECTRONICS

PHASE II

FINAL REPORT

9 APRIL 1979

Work performed under Contract No. E(04-3)-1309

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Abstract

This report describes the second year (1978) of development on the Mechanical Refrigerator for Deep Well Sonde Electronics. The system is based on a vapor cycle refrigeration system with the usual compressor, evaporator and condenser. It is designed to operate between a condensing temperature of 275°C and an evaporating temperature of 125°C. Water vapor is the refrigerant. The compressor is an electric motor driven three-stage positive displacement machine. The design output of the refrigerator is 50 watts cooling. The work described in report covers the development of the compressor with associated component development tests.

The compressor was tested in a loop simulating downhole conditions. Short duration runs were accomplished where steady state conditions of refrigerant and lubricant flows were established. However, because of various component failures, sustained operations, greater than 6 hours at a time, could not be made. (The design goal was 100 hours.)

Based on the results of these tests it appears that considerable more development work, particularly at the compressor component level are required before the 100 hr design goal can be met. Recommendations for such development work is included.

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1.0 INTRODUCTION

This program is a continuation of Contract E (04-3)-1309 for the development of a compressor for a vapor cycle refrigeration system. The refrigeration system is to be used to cool instrumentation in a downhole sonde. This approach permits the use of existing well logging electronics for prolonged periods downhole. Phase I, completed in June 1977, resulted in the partial development of a three-stage reciprocating compressor capable of developing usable flows and pressures in a thermal/steam environment.

The objective of Phase II was to continue the development up to the point of delivering a working, sonde-adaptable refrigerator. The Phase II program had four major work tasks. These were (1) compressor development, (2) controls development, (3) drive systems procurement and (4) system design and integration. A number of significant successes were realized that demonstrate the potential of this refrigerator. The basic high temperature performance of the compressor and its lubrication system was successfully demonstrated. The major control valve was successfully developed. Supplies for the high temperature drive motor were located. Several compressor reliability problems remain to be solved though, in order to attain a long operating time at high temperatures. The details of the results and the work performed are discussed in subsequent paragraphs.

1.1 REFRIGERATOR DESIGN OBJECTIVES

The objective of the Phase II program was to continue the development of a deep well refrigerator compressor started in Phase I and, in addition, developed the associated drives, controls and packaging. The basic design requirements are described in Section 2.3 and summarized below:

- o Temperature up to 275°C
- o Downhole pressure - 7500 psi
- o Cooling capacity - 50 watts at 125°C
- o Operating Time - 100 hours

- o Input power - 3 phase, 60 cycle AC
- o Envelope - 3-5/8" OD, length TBD

1.2 SUMMARY OF RESULTS

1. The Phase I compressor was redesigned and tested. The results of these tests showed that, although lubricant and steam flow were established, deficiencies in the valving and seals limited operation at high temperature to two to four hours.

2. As a result of the Phase I compressor experience, an improved compressor incorporating new seals and valves was designed and tested. Short duration (six hours) continuous operation up to 537°F (275°C) was achieved. However, due to poor component reliability, or possibly wear, consistent long term operation could not be achieved.

3. The most serious deficiency was the poor volumetric efficiency at temperature. Possible causes are worn rings or excessive valve leakage.

4. The pressure regulated liquid expansion valve, a major control item was demonstrated at temperature.

5. A specification was developed and bids were solicited for a high temperature motor suitable for downhole use. The bids of \$34K and \$60K were put "on hold" due to the lack of progress in the compressor development.

6. A detailed design of a refrigeration system integrated into a downhole sonde was accomplished and is included with this report.

The remainder of this section (1.0) expands upon the summary statements above. Section 2.0 gives the program background including a summary of the (1976-1977) Phase I effort. Section 3.0 describes the compressor development. It includes a discussion of the development work performed on

valves and seals. Section 4.0 describes the control system components development. Section 5.0 covers the drive motor specification and vendor liaison activity. Section 6.0 describes how the system would be packaged as an integrated sonde.

1.2.1 Compressor Development

Work performed in the first half of the development followed the proposed task outline including the following items:

- a. Reconfigure valve geometry to improve stage 3 suction valve performance.
- b. Enlarge oil sump and relocate oil pump to ensure continuous steady state lubrication.
- c. Add steam-oil separator to aid oil recycle capability.
- d. Refurbish test system to improve thermal control.
- e. Test all-up compressor system at temperature.

Proposed work was performed, culminating in all-up system test in May 1978. The results showed that the system was operable, with lubricant and steam recirculation achieved. However, component reliability and system operability were so poor a new compressor design was started. The principal deficiencies were seals and valves.

The new compressor design incorporated results of parallel component development tasks on:

- a. static flange seals,
- b. dynamic shaft seal,

- c. tubing feed-throughs, and
- d. discharge ball checks.

The modified compressor was subjected to an all-up system test with substantially more success than the earlier tests, but it still fell short of the design goal. The first run lasted six hours versus a design goal of 100 hours.

Satisfactory operation, i.e., no failures, was obtained from the following components:

- Static flange seals, using Viton o-rings.
- Pressure lubricated crank and rod bearings. (The crank surface was flame sprayed Triballoy 100. The bearing material was bearing bronze.)
- Piston wrist pins - splash lubricated
- Brass oil pump gears
- O-ring (Viton) sealed tubing feed throughs.

Major problems still unsolved are:

- a. Poor flow and pressure stability
- b. Suction valving and shaft seal life and/or reliability

No conclusions have been made on the cause of these problems other than the lack of maturity of the basic design.

1.2.2 Control System Development

System tests have demonstrated that the only component required for continuous operation is a suction pressure regulated expansion valve. A modified all-metal pressure regulator (Consolidated Controls #2338 High Purity Gas

Regulator) controls flow and pressure satisfactorily at temperature. Start-up techniques after prolonged shutdown have not yet been developed. Short duration (minutes) shutdowns can be tolerated because the machine is capable of restarting with only a marginal increase in torque. Long duration (hours) shutdown start-up solutions depend upon an absolute "zero leak" isolation valve between the low and high pressure sides of the system or a discharge bypass. Vendor surveys, original design work and laboratory tests have supported this investigation. Of these, designs based on a zero-leak check valve look most promising.

The use of high temperature DC solenoid-driven valves has been abandoned because of (1) vendor disinterest and (2) complications of the DC control system.

1.2.3 Drive System

A drive system specification was developed, vendor contacts made, and bids received. The basic requirements call for a 1/4 HP, 3 phase, 60 cycle machine, with a geared output shaft running at 3000rpm. The two bidders responding were:

American Electronics - \$34K
Fullerton, California

Skurka Engineering - \$60K
Los Angeles, California

A key element in the high temperature motor design is the choice of the winding insulation. American Electronics proposed a ceramic fiber insulation, while Skurka proposed a high temperature enamel. Skurka's higher bid apparently reflects their uncertainty on the gear box and its lube requirements. Since issuance of that specification, redefining the interface by incorporating the gear box in the compressor rather than the motor has been considered.

The procurement has not been made and is now "on hold" until development problems incurred on the compressor are resolved.

1.2.4 System Design and Integration

A detailed design and analysis of a downhole refrigerator was accomplished. The design was based on past and concurrent theoretical and experimental data developed in Task 1. Key features of the machine were:

1. Proportions of pistons, crank, oil pump are identical to 1977 Phase I machine.
2. Static flange and tubing feed-through sealing accomplished with o-rings (Kalrez or Viton).
3. Overall unit sealed with Inconel shell, 3 5/8" OD x 1/16" wall, brazed to assembly top and bottom.
4. Dynamic shaft sealing by lapped metal-to-graphite running surface.
5. Minimum controls - uses only suction pressure regulator.

The all-up system was not fabricated. However, the compressor block, with drive adapter and modified oil sump, was fabricated and subjected to test.

1.3 RECOMMENDATIONS

Many of the problems encountered in this development may have been the result of a somewhat over-optimistic view of the difficulty of the problem. This view led to the commitment of an ambitious program where a complete Sonde cooling system was to be developed in one step. This large task hindered our efforts to concentrate on the few truly critical problems. Accordingly the tenor of the recommendations that follow suggest a "back-to-basics" course where component level development is emphasized.

- a. Future, near-term, work should be concentrated on the compressor.
- b. Compressor component level development should continue on the following elements:
 - o Suction valves - This should include a design study of the block porting. The existing passages may hinder block-to-head sealing because of the narrow lands between ports.
 - o Piston & Piston Rings - Recent investigations in this area indicate that a potential source exists for high pressure, miniature pistons and rings.
 - o Shaft Seal - Continue development and life tests on shaft seals. Use of improved decoupling bellows to be investigated.
- c. Motor development should be considered a separate problem (or procurement). Furthermore, since there appears to be a need for a fractional horsepower high temperature motor, this motor should be designed as a general purpose motor without a gear box. A 3-phase induction motor appears to be a good choice for a design basis.

2.0 BACKGROUND

A description of the Phase I development effort is presented. Phase I was concerned with the development of a positive displacement, reciprocating compressor suitable for use in a downhole sonde cooling refrigerator.

2.1 PHASE I SUMMARY

The program objective was development of a water vapor cycle refrigeration compressor. The refrigeration cycle was designed to operate between a sink temperature of 125°C (257°F) and a source temperature of 275°C (527°F) such as might be encountered in a downhole instrumentation sonde. The refrigeration cycle was designed to provide up to 50 watts effective cooling at a baseplate temperature of 125°C.

Envelope dimensions for the compressor section were set at 3.0 inch OD. The length was not restricted. The Phase I machine was not designed to take geothermal well pressures, but was intended to operate in a thermal environment of at least 275°C.

Phase I development was limited to the compressor as it was considered the high-risk element. The drive system, controls, and refrigeration components were slated for later development.

Work performed included thermodynamic analysis, compressor design, fabrication, and testing. Results from Phase I were somewhat mixed. The compressor did indeed meet the basic pressure and flow requirements with air at room temperature conditions. Tests with steam up to 500°F showed that the machine could indeed compress and flow water vapor at the required rates; but operations could not be sustained for more than an hour because of test fixture limitations and component failures.

The following paragraphs describe the basic concept, compressor, test results, and the follow-on Phase II program plan.

2.2 CONCEPT DESCRIPTION

The downhole refrigerator design is based on a straight-forward application of a vapor cycle refrigeration concept. Figure 2-1 illustrates the key features of such a vapor cycle refrigerator.

Figure 2-1. Vapor Cycle Refrigeration Concept

The compressor is a device used to raise the pressure of vapor from the evaporator high enough to condense to a liquid in the condenser. Once converted to a liquid, the fluid is flashed to a vapor in the evaporator where it absorbs heat from the payload. In essence, the refrigeration system is a "heat pump" which lifts heat from the low temperature evaporator and rejects heat at high temperature in the condenser.

Water was selected for the working fluid as a result of Phase I studies. The choice was based on its good thermodynamic properties, availability, safety and well-established thermal properties. Other refrigerants were considered

but discarded. Conventional refrigerants, such as the Freons have critical pressures far below 2750C and hence will not operate in a vapor cycle system. Unconventional fluids such as toluene and chlorobenzene offer lower condensing pressures than water, but require large compressor throughputs.

The diagram above represents a primitive system. A significant improvement in cycle efficiency can be made with some relatively simple plumbing changes. These are shown below:

Figure 2-2. Modified Vapor Cycle Refrigeration

A liquid subcooler and interstage intercoolers have been added. The liquid subcooler utilizes the still cool vapor exiting from the evaporator to cool

the hot inlet high pressure liquid. The intercoolers permit the compressor to operate more nearly isothermal (the lowest energy mode).

The thermodynamic state points for such a refrigeration system are shown below on a conventional pressure-enthalpy plot:

Figure 2-3. Refrigerator Thermodynamic Cycle

The analysis associated with this plot is appended. The primary result is that 750 Btu of cooling effect can be extracted per pound of steam circulated. Therefore, to accomplish 100 watts of (gross) cooling, a steam flow of 0.0075 lb/min is required.

The compressor meeting this basic flow requirement developed in Phase I is described further below.

Figure 2-4 shows the key elements of the compressor. The cast-iron block is nominally 3 in OD x 6 in. long.

Figure 2-4. Compressor Key Elements

2-5

The nominal cylinder/crank dimensions (in inches) are:

Stage 1 - .625" Bore x .40" Stroke

Stage 2 - .375" Bore x .40" Stroke

Stage 3 - .219" Bore x .40" Stroke

The crank throws are at 120 degrees with a "firing order" of 1, 3, 2.

Each steel piston was fitted with a pair of diagonally slotted aluminum-bronze piston rings having a rectangular cross-section (.025 x .020). They were installed with a "line-to-line" fit to the bore.

The 17-4PH crank was plasma flame coated with Dupont's Triballoy, a hard surfacing bearing material. The mains ride directly on cast iron. The crank pin bearings are bronze.

The oil pump is a direct drive gear pump (.250 PD x 24 diametrical pitch brass gears). It draws oil up from a three ounce capacity sump.

The suction flow is controlled by 0.004 s/s chem-milled reed valves. The discharge valves are spring-loaded ball checks.

The performance of the compressor during Phase I showed that it was capable of meeting flow and pressure requirements. This was based on tests primarily using ambient air and a few tests with steam. The steam tests, although not conclusive, did show that the compressor could be operated with steam without noticeable wear or damage.

Test points in Figure 2.5 taken from the Phase I test report, show the compressor performance characteristics. Volumetric efficiencies of 60-65% were obtained. This level of efficiency assured us that sufficient flow could be produced to provide at least 50 watts of cooling assuming, of course, that steam behaved similar to air at the design pressures. The steam test points, scattered as they were, indicated that steam would behave similar to air. (The scatter was due to the difficulty in maintaining steady state test conditions.)

Figure 2-5. Effect of Speed and Suction Pressure
(Taken from Phase I test report)

2.3 PHASE II PROGRAM PLAN

As Phase I came to a close, it became apparent that major deficiencies still existed, preventing a full demonstration of the compressor as a reliable long duration machine.

The major deficiencies uncovered in the development testing were:

- a. Oil loss through the discharge, limiting operating time.
- b. Ball checks did not shut tightly, which affected volumetric efficiency at high pressure.
- c. Stage 3 suction valve reed did not appear to seal properly.
- d. The test fixturing allowed so much heat loss that closed loop recirculation could not be obtained.

Because the work performed in Phase I showed promise, a follow-on Phase II program was funded by DOE. The proposed work was expanded to include the development of an entire sonde cooling system. The key tasks in this program are listed below.

- Compressor Development
- Refrigerator Controls Design and Fabrication
- Drive System Design and Fabrication
- System Integration
- System Thermal-Pressure Test

New ground rules were also established for the system. These are repeated on Table 2-1.

One significant difference between Table 2-1 and the original requirement is the increase in allowable body diameter from 3 inches to 3-5/8 inches. This allowance makes packaging considerably easier.

The sections that follow describe the work performed on each of the Phase II tasks.

Table 2-1. Basic Design Performance Requirements for a Downhole Refrigeration System For Cooling Geothermal Well Logging Tools.

OPERATING DOWNHOLE ENVIRONMENT

- Temperature - Up to 275 C (527 F)
- Pressure - Up to 7500 psi
- Media - Highly corrosive geothermal vapor or fluid containing large amounts brine with hydrogen sulfide, methane, etc.
- Depth - Up to 15,000 feet

COOLING CAPACITY

50 watts (minimum) delivered to electronics package

MAXIMUM ELECTRONICS PACKAGE OPERATING TEMPERATURE

125 C (257 F)

MINIMUM OPERATING TIME

100 hours

INPUT POWER

3 phase, 60 Hertz power at a voltage to be specified

INPUT CONTROL

The system is to be designed to require no external control other than an on-off switch at the surface. Power and control will be limited to 4 conductors, one of which will preferably be the cable armor.

ENVELOPE

- Diameter - not to exceed 3-5/8 inches (current design is 3")
- Length - 4 to 6 feet (as required by final design)

INTERFACES

Cable (top) end. The cable end interface will consist of a metal gasketed bolted flange. A cylindrical cavity below the flange will be provided to permit field splices as required. The details of this flange will be coordinated with DOE so that a standard cable head can be used. A special flange test adaptor will be provided for the proof test, to interface with the lab autoclave. A cable head is not provided in this contract.

Payload (Bottom End). The payload interface will be metal gasketed. Space will be provided for the refrigerant inlet and outlet lines, as well as the payload instrumentation cables.

Instrumentation Cable Pass-Thru. A passage having a cross section area equal to a 1/2 inch hole will be provided for instrumentation cables through the refrigerator package. A portion of this passage such as through the motor section will be a flat oval, typically 1/8" thick. SDC's position at this time is that high temperature electrical feedthroughs be permanently installed with only the connections being made up in the field. Attempting to "fish" through special cables in the field is at best hazardous and could reduce reliability. Potential requirements such as fiber optics cables are best pre-installed and tested in a shop.

3.0 COMPRESSOR DEVELOPMENT

The work planned in this task was to complete the development of the Phase I compressor to a point where it's operability could be demonstrated in a continuous 100-hour run. Subtasks included:

1. Redesign of suction valving, lube system and test fixturing.
2. Rework of compressor internals such as piston rings and bearings.
3. Fabrication of test fixturing to permit continuous closed loop operation at temperature.
4. Tests to demonstrate redesigned/reworked compressor would be operable for 100 hours.

As work progressed towards accomplishing these tasks it became obvious that the existing design was inadequate. Midway through the program additional effort was expended to improve component performance and reliability. This latter effort was followed by another series of all-up system tests.

For purposes of presentation the first half of this development program is described in Section 3.1, Compressor Development - Part A, and the latter half in Section 3.2, Compressor Development - Part B.

3.1 COMPRESSOR DEVELOPMENT - PART A

3.1.1 Compressor Redesign and Rework

No major changes in the basic Phase I design philosophy were made. The major compressor redesign-rework elements are shown in Figure 3.1.

Figure 3-1. Compressor Redesign Elements

The motor-test-fixture adapter was included to permit better alignment between the drive motor and the compressor. It also served to align the compressor assembly with the test chamber.

The lube system was redesigned to include a large integral sump (26 cubic inches). The oil pump previously located in the block was relocated to the bottom of the oil sump. No screens or filters were included in the Part A design. The oil sump was equipped with four 1/8 NPT pipe taps to permit access as well as for measurements of oil level.

Two suction valve modifications were made to the stage 3 reed. These changes were designed to improve the structural reliability. The original (Phase I) reeds were weakened by an excessively large discharge port hole. They are shown sketched below:

Figure 3-2. Stage 3 Reed Modifications

Three samples of each design were chem-milled in .004 and .006 inch thick full hard 304 stainless steel. The chem-milling was subcontracted to:

Chemical Machining Corp.
142 Nevada Street
El Segundo, CA 90245

New piston rings (2 per piston) were fitted. The rings (having a nominal .025x.020 cross section) are made from spring tempered aluminum bronze. They were hand fitted to the cylinder bores line-to-line with a nominal 0.030 end gap.

Based on various literature references, the possibility of making the rings with cross sections other than rectangular was investigated. However, no further attempts were made to implement these designs, mainly due to their small size.

3.1.2 Test Fixture

An electrically heated thermal chamber was designed and fabricated. It is essentially a 5 foot long x 6 inch diameter steel pipe with flanged ends. The bottom flange is blind and serves as a partial support. The upper flange mates with the motor-compressor adapter. A pair of 1-1/2 NPT pipe nipples welded slightly above the midpoint of the pipe act as pivot supports for the test chamber. An angle iron (2 x 2 x 1/8) frame supports the test chamber. The entire assembly is placed in a sheet metal pan to catch hot oil in case of an accident. A sketch of the test chamber appears in Figure 3-3.

The heat transfer medium inside the chamber is a high temperature synthetic -- Monsanto Therminol 66. Heat is supplied by four (4) 1 KW band heaters attached to the outside of the pipes. Two heaters are located near the bottom, one in the middle and one near the top. Temperature control is accomplished by manually adjusted Variacs, one for each heater. Feedback is provided by thermocouples welded to the pipe near each band heater. Approximately 6 inches of silica fiber insulation is wrapped around the pipe.

Figure 3-3. Test Fixture

No particular problems were encountered in using this test chamber. Temperature control, although somewhat tedious, was not difficult. The four band heaters permitted reasonable control over the axial temperature distribution within the chamber. Generally, temperatures could be held within 5°F top-to-bottom. Considerable amounts of oil vapors vented from the chamber during operation. As a consequence, the test fixture was placed under a large (6 x 6 foot) vent hood exhausting to the outside. The vapor, which Monsanto claims is not toxic, is somewhat unpleasant and tends to linger long after the test.

Because of exposed thermocouple leads, the dielectric strength of Therminol 66 was questioned. Bench tests showed that this was not a problem. Conductors one centimeter apart in 550°F oil have a resistance in excess of 100 megohms.

The drive is a size 34 Bodine motor (1/3 HP, 5000 RPM). It is normally used with a Bodine supplied SCR controller. Speed is measured with an Airpax 14-0001 speed pickup sensing a 19 tooth sprocket attached to the motor shaft.

The existing Bodine controller was modified with stabilizing circuitry so that the drive motor behaved like a shunt wound D.C. motor. It appears to have a 10-15 percent droop from no-load to full load.

An earlier attempt to make a motorized feed-back loop was abandoned because of uncontrollable potentiometer and motor slop.

Except for an occasional power supply overload, the motor control system worked quite well.

3.1.3 Development Tests

The rework tasks described in Section 3.1.1 were accomplished and then subjected to room temperature tests with air. The results were disappointing in that no significant improvements in performance were measured. Volumetric

efficiency was nominally 70 percent at a discharge pressure of 900 psig which is comparable to the better runs made during Phase I.

The lube system relocation did, however, make operation much easier since the oil pump would prime itself automatically.

The suction valves were a disappointment. Neither Mod 1 or Mod 2 (Section 3.1.1) performed much better than the original Phase I design. We did note, however, that the thinner (.004) reeds appeared to seal better than the 0.006 inch thick reeds.

The flanges were sealed using a silicone RTV. The valve head and reed assembly was gasketed with .001 thick aluminum foil. An attempt to use chem-milled copper foil (.002 thick) was not successful. The shaft seal was made with a spring loaded, graphite filled teflon unit (Balseal 50B-10-MSGC). Except for difficulty in making the head seal, no other problems were encountered with sealing at room temperature.

3.1.4 Thermal System Tests

After what was considered satisfactory room temperature performance runs (70 percent volumetric efficiency to 900 psig), the compressor was integrated into an all-up refrigerator configuration. Principal elements making up the system were:

- Intercoolers -
 - Stage 1-2 - 10 feet 1/8" tubing
 - Stage 2-3 - 11 feet 1/8" tubing
- Condenser - 120 feet 1/8" tubing
- Receiver - 14.3 cu-in bottle
- Expansion Valve - Consolidated Controls #2338 Gas Regulator
- Evaporator - 10 feet 1/8" tubing
- Shut-off & Bypass Valves to permit compressor start-up & flow metering

The expansion valve, evaporator and manual controls were located outside of the main test chamber. They were separately enclosed and electrically heated. The compressor, intercoolers, condenser and liquid receiver were placed in the test chamber filled with Monsanto Therminol 66. Figure 3-19 (page 3-32) shows the general arrangement of the system elements.

The flow schematic and the instrumentation sense points are shown on Figure 3-18 (page 3-31). The tests conducted in Part A did not include all pressure transducers as shown on Figure 3-18 and hence relied largely on Bourdon tube display gages for operating condition.

Several all-up systems tests were attempted with some success in May and June 1978. The first, on May 10, was to determine its operability. It was a one-hour run at a temperature range from 450 to 470°F. A condensing pressure of 460 psig was reached. The compressor operation was quite smooth with steady interstage pressures. The following day, May 11, a three-hour run was accomplished with the same configuration as on May 10. It was aborted at a chamber temperature of 500°F when a sudden interstage pressure rise caused an increase in the drive torque. A post-test examination showed that leaks had developed in all the flanges sealed with RTV. The shaft seal had also failed.

Since the May 10-11 tests were shut down before meaningful flow data could be obtained, further tests were attempted through May and into June. Finally, on June 17 a one-hour test was performed up to a temperature of 490°F. Flow measured during this run using weigh tanks was 0.0075 lb/min. This test was aborted due to a failure in the drive motor power supply system. No further system tests were performed on the Part A compressor.

3.2 COMPRESSOR DEVELOPMENT - PART B

After the Part A system test failures it became apparent the compressor was not adequately designed for operation at high temperature. Therefore, further attempts at implementing an all-up system test were cancelled in favor of developing high temperature sealing techniques and improved valving. The

following paragraphs describe the component levels tests in this effort together with the final system tests incorporating the results of these component development tasks.

3.2.1 Static Flange Sealing

High temperature O-rings and precision flat lapping were the principal techniques used to effect static seals. A seal test fixture was designed and fabricated to determine the relative effectiveness of various O-rings and lapping techniques. This fixture is shown on Drawing TNT-82 (Figure 3-4). It consists of two 17-4PH 3.50 OD disks with lapped surfaces and/or O-ring grooves. The fixture was also configured to accept tubing feed through tests. The mating flanges were clamped with 12 4-40 screws. The net clamping surface area per screw was nominally 0.31 square inches. Assuming a bolt stress of 100 ksi, the unit clamping pressure is 2000 psi. Standard O-ring groove dimensions were used. No special allowance for temperature was made.

The first tests were with Dupont Kalrez O-rings having 1/16 inch cross sections. The tests were conducted by assembling the units with the internal cavity partially filled with water and lube oil (Brayco 815Z). The assembly was then heated, to 550°F (289°C). A 3-1/2 inch band heater was used. Thermocouples were used to measure test fixture temperature. Leakage was detected both visually and with a pressure gage sensing cavity pressure. The following results were noted:

1. Kalrez O-rings effectively seal the flange pairs, i.e., no leaks after 3 hours, at temperatures from 70°F to 550°F.
2. Disassembly shows considerable O-ring flash indicating extrusion.

Figure 3-4. Drawing TNT28

3. A used O-ring will frequently leak until the temperature is 200°F or better, after which it remains tight.
4. Tubing feed-throughs can be made with O-rings, provided the O-ring is "tight" around tubing. For example, a size 007 O-ring (.145 I.D.) will not seal around a 1/8 tube. A size 006 (.114 I.D.) will seal.

These tests, conducted early in a Part B effort, were performed primarily with Kalrez O-rings because it was (mistakenly) felt that the lower cost Viton O-rings were only good to 260°C (500°F). Later in the program, however, similar tests showed that Viton was at least as good as Kalrez up to 550°F. As a result, the compressor seals thereafter were made with Viton O-rings.

Flat lapped surfaces are marginally effective. Part of this may be due to the great care which must be taken in handling surfaces finished to 1 μ -in. Small amounts of contaminants or scratches can easily affect the sealing effectiveness. As an example, the compressor head-to-block surface is lapped. The first time the head was installed sealing was easily effected. After repeated installations it became necessary to resort to aluminum foil gaskets.

The Micro Surface Engineering Co. of Los Angeles provided the lapping services on this program.

3.2.2 Shaft Seal Development

Since the compressor has an external drive, a dynamic seal must be provided between the shaft and the crankcase. The crankcase pressure is 50 to 100 psi above ambient. The Phase I machine was equipped with a Teflon lip seal from the Balseal Co. of Costa Mesa, California. A typical lip seal installation is shown on Figure 3-5.

Figure 3-5. Typical Lip Seal Installation

These seals work quite well. They are simple to install and quite tolerant to shaft misalignments and normal vibration. For our applications, however, they seemed to be limited to temperatures below 450°F despite high temperatures claimed for graphite filled Teflon units.

As a result of the lip seal failure, a development effort for a shaft seal was initiated. The requirements for the seal were tentatively set at:

Temperature	- 600°F
Pressure Difference	- 100 psi
Shaft Diameter	- .25 in.
Speed	- 3000 RPM

Originally, a survey of existing shaft seals fitting these requirements was made. The most promising lead was from the John Crane Packing Co. They offered a 1/4 inch shaft seal with a polished metal-graphite seal face. However, after a month we were informed that the 1/4 inch unit was not stock and that some development would be required before delivery could be made. At that point we dropped the commercial seal and proceeded with our own development.

Our first attempt, shown in Figure 3-6, would seal only at room temperature. As soon as the temperature was increased the oil would thin out and leak.

Figure 3-6. Shaft Seal, Version 1

Various attempts were made to improve sealing by improving the surface finish on the seal faces. One attempt included making the shoulder separate, allowing it to be polished before it was pressed on the shaft. This improved sealing only marginally.

At that point it was decided to mechanically decouple the seal face from the graphite plug. This was done by supporting the shoulder on a bellows as is shown below:

Figure 3-7. Decoupled Shaft Seal
3-13

The first version of this seal, used in the September 19 test, was assembled with high temperature epoxy (Duralco 700). The bellows is brass and has 10 convolutions, 1/2 OD x 5/16 ID. The seal faces were polished to a 1 -in finish. The seal appeared to be effective throughout the six-hour test (no oil leaks were noted), however, the epoxy bond failed shortly after test shutdown. Figure 3-9, next page shows a cutaway view of the seal assembly.

A second version of this seal was installed for the October 12 test. In this version the bellows was silver brazed rather than epoxied. The October 12 test was not shutdown due to a shaft seal related failure however, a post-test examination showed that the bellows itself had developed a crack.

The final version of the seal was made with a formal stainless steel bellows, #55320-30-74, Miniflex Corp., Van Nuys, California. It has 7 convolutions 5/16 ID X 1/2 OD. This stainless steel bellows was considerably stiffer than the brass unit it replaced. Static & dynamic tests however, show that it sealed "bubble" tight. This unit survived the November 1 test but failed the November 15 test. The bellows was found fractured. Based on this experience, it would appear that a more compliant edge-welded bellows would be better than a formed bellows.

3.2.3 Discharge Valves

Effective suction and discharge valves appear to be key items to the success of the machine. Developing effective discharge valves was the most difficult and time consuming of all component development tasks. The original Phase I machine had discharge ball checks which were integral with the head. This concept is shown below.

Figure 3-8. Phase 1 Discharge Valve

COMPRESSOR DRIVE SHAFT BEARING & FACE SEAL ASSEMBLY

Figure 3-9. Compressor Drive Shaft Bearing & Face Seal Assembly

In the earlier versions, the ball seat was formed by a milling cutter with a 120 degree conical end. The seat was finished by a combination of coining and hand polishing. The results were mixed. On some occasions a fairly effective seal could be made. Generally the results were undependable partially because the seat, being recessed, was difficult to inspect.

As a consequence of the difficulty encountered in the Phase I design, a concept described in the literature and recommended by valve manufacturers was tried. This design referred to as the "tri-point" seat is shown sketched below.

Figure 3-10. Discharge Valve Tri-point Seat

It is called a tri-point seat because it is formed by the intersection of three radii. The final seat lapped to actual ball diameter is quite narrow so that a high contact stress is obtained. Data shown in an Air Force sponsored study,* show contact stresses as high as 3000 psi.

Our experience with this seat form was poor. These seats, formed for us by Micro Surface Engineering, Los Angeles, simply would not seal under compressor

*Air Force Technical Report AFRPL-TR-66-147, "Poppet and Seat Design Data For Aerospace Valves," July 1966, Rocket Propulsion Laboratory, Air Force Systems Command, Edwards AFB California.

operating conditions. The seats were very subject to contaminant damage. The measured leakage rates represented a significant fraction of the compressor flow.

Because of the difficulty in making effective metal-to-metal seals, a soft seat design utilizing Kalrez and Viton O-rings was developed. This concept is shown below.

Figure 3-11. Soft Seat Design

This design provides bubble-tight sealing and is easy to implement provided operation is limited to room temperature. A problem arises when the valve assembly is operated hot. The O-ring expands, distorts and frequently completely plugs the hole. The solution for this thermal problem was to essentially mold the seat in place at temperature. This was accomplished by a special tool inserted in place of the ball and literally cooking the assembly in an oven. This tool forced the O-ring into the corners of the groove and caused excess material to extrude through the bottom of the hole.

The final assemblies were made with Viton O-rings cooked to about 550°F. Viton appeared to provide a smoother finish than Kalrez. Size 003 O-rings were used.

The September 19 System test was run using these molded Viton discharge valve seats. Performance through the first half of the test was excellent, torque was low, and interstage pressures were steady and also low. However, midway through the test, torque and interstage pressure oscillations were noted. A post test diagnosis indicated that the Stage 3 valve seat had failed. (The ball had extruded past the O-ring seat.) Stage 1 and 2 O-ring seats did not appear to be damaged. This experience convinced us that "soft" valve seats required major advances in material technology so it was back to the metal seats.

The discharge valve seat finally used was a variation of the original Phase I ball checks. These ball checks use .140 diameter grade 25 balls in 30 degree coined conical seats per the sketch below:

Figure 3-12. Ball Check Discharge Valve

These valves still leak, but at rates which seem tolerable. Figure 3-13 shows a plot of leakage rates for various seat geometries. Experience to date indicates that leak rates in excess of about 10% of nominal compressor flow will measurably affect performance. Leakage rates as low as five percent of compressor flow were achieved.

3.2.4 Suction Reed Valves

Thin flexible metal reeds are used to check suction side pressure flow. The original concept was an integral chemical milled valve plate sandwiched

Figure 3-13. Effect of Seat Angle on Leak Rate

between the head and block. The two problems with this design were (1) the difficulty in effecting a leak tight head-to-block seal and (2) the cost and time involved in chemical milling. The latter difficulty inhibited the investigation of many designs.

As a result of the problems on the original machine, single reed assemblies were made for each stage. These assemblies were trapped in a cavity between the block and head and therefore did not affect the flange sealing. Flange sealing was accomplished by using a large number of screws and lapped surfaces.

In our current test machine, the valves reeds were hand made and attached by spot welding. Photographs of the installation are shown below. The final machine would have chemical milled reeds. Sample configurations are shown on the next page, Figure 3-15. Performance did not seem to be strongly affected by reed valve design as long as it wasn't grossly bent or otherwise distorted. Data from our most recent tests however indicate that some other phenomena such as contamination or lack of a lubricant film deleteriously affects performance.

Figure 3-14. Photo - Suction Valve Reed Assembly
3-21

Figure 3-15. Sample Stage 3 Reed Valve Configurations

3.2.5 Compressor Block Assembly

Midway through the program a new compressor block was designed and fabricated. A sketch of the new block assembly is shown on Figure 3-16. The new block was designed to take advantage of the results of the component tests described earlier. The diameter was also increased to 3-1/2 in OD. which made packaging easier. The block was made of centrifugally cast iron normalized to 30 ksi. The Phase I block was used as-cast and was difficult to machine because of residual carbide.

Considerable care was taken to insure easy and positive sealing. The head surface was completely flat so that it could be lapped. The original design, see Figure 2-4, provided a recess for the head attachment. This recess prevented flat lapping. O-ring seals were provided for the crankcase cover, shaft seal attachment, tubing feed-throughs and adapter and oil pump flanges. Standard O-ring groove dimensions were used, i.e., no allowance was made for temperature. Both Kalrez and Viton O-rings were used. No modifications were made to the crank or pistons. The pistons were, however, refitted with new aluminum bronze piston rings. The lube system remained essentially as before except for an oil supply for the shaft seal assembly. The oil pump gears were also remade with a closer tolerance.

Two adapter plates were added to permit the new 3.50 OD block to interface with the existing 3.00 inch motor adapter and the oil sump.

The valve and head design was substantially changed from the Phase I machine. In the original machine the suction valves were formed by a single chem-milled sheet, sandwiched between the head and block. This design made sealing difficult because the valve sheet was directly in the clamping-leak path. The discharge valves in the original machine were integral with the head. The valve seats therefore were difficult to access and inspect. For these reasons the new design was made with a two piece head with individual suction reed located in a recess, out of the clamp-leak path. Photographs of the head block assembly are shown on the pages that follow (Figures 3-16 through 3-19).

Figure 3-16. Block Assembly

The head was made from rectangular blocks of 17-4PH to save cost. The final machine would be machined to the contours of the cylindrical compressor block.

The valve head and valve plates were flat lapped to 1 μ -in to ensure leak tight sealing. A major advantage of this two-piece construction was that it allowed us to inspect or change the discharge ball check seat easily. The main difficulty encountered was with the installation suction reeds, particularly on Stage 3. In this design concept, the suction reeds were to have been formed by chem-milling. However, because of time and cost considerations, hand made reeds were installed. These reeds were spot welded in place with a small re-inforcing strap to maintain closing tension. This was an extremely tedious, difficult to control procedure due to the small size, particularly in Stage 3. For example, heat from spot welding frequently caused the adjacent ball check valve seat to distort and start leaking excessively. Clearly this procedure is not suitable for a finished machine. Chem-milled reeds trapped or partially clamped between the head and block appears to be the way to go. Photographs of the new assembly and components described above follow (Figures 3-17, 3-18, 3-19).

Figure 3-17. Compressor Assembly

Figure 3-18. Crank and Piston Assembly

Figure 3-19. Valve and Head Assembly

3.2.6 Checkout

The first room temperature test was a disappointment. The flow rate appeared to be about half of what was expected. An investigation of the potential problems indicated that Stage 1 was being starved by a suction port that was too small. (It had been reduced from .16 in. diam. to .08 in. diam. on the premise that at the design flow rates the pressure drop would be negligible.)

Flow returned to normal after the Stage 1 suction port was increased to .125 in. diam. Volumetric efficiencies between 70 and 75 percent were consistently obtained after this fix with room temperature air.

The discharge check valve seats were made with size 003 Viton O-rings. Static leak checks made on these valves show that they seal effectively "bubble tight."

A separate experiment was conducted to determine the effect of piston ring blowby. A special Stage 3 piston was made and fitted with a size 010 (1/4 OD) Viton O-ring. Static leak checks on this design showed that it sealed bubble tight. This compares with a leakage rate 130 cu. in/min (at 100 psi) for the normal rings on Stage 3 piston.

Dynamic performance, however, did not seem to be strongly affected by piston blowby (measured statically). Compressor volumetric flow rates up to 900 psi were very nearly identical for either configuration with room temperature air. In retrospect, this data may not extrapolate well to high temperature operation. It appears that much of the dynamic sealing is accomplished by the presence of generous amounts of viscous oil. It is possible at higher temperatures when the oil thins out that "hydrodynamic" sealing is not present. Some of our high temperature runs indicate this.

A significant improvement in the oil pump system was noted. Discharge pressures over 300 psi were observed when the machine was cold. Flow also increased from a nominal 6 cu-in/min, measured in the Part A machine, to 11 cu-in/min. (The theoretical displacement of the oil pump at 3000RPM is 18 cu-in/min.)

3.3 SYSTEM TEST PROGRAM

The purpose of the system test program was to demonstrate the operability of the compressor-refrigerator system at temperature.

Work performed in the system test program included compressor operation and calibration at room temperature followed by tests at temperature in a thermal chamber. The following paragraphs describe the test procedures, set-up and results from the system tests.

3.3.1 Test Plan

The test item is a three-stage, positive displacement compressor plumbed to simulate a downhole instrument cooling refrigerator. The compressor is externally driven by a torque-flexure restrained electric motor running at constant speed (3000 RPM).

The refrigeration simulation consists of intercoolers, condenser, liquid receiver, expansion/regulator valve and evaporator. The compressor and associated high pressure plumbing are contained in a temperature controlled oil bath. The expansion valve/regulator and evaporator are external to the oil bath and the temperature controlled separately.

The system is instrumented so that critical performance parameters may be measured. A schematic of the test system follows, together with a general test set assembly (Figures 3-20 and 3-21).

Test Objective - To demonstrate the operability and determine the performance characteristics of the compressor/refrigerator system at the design temperature of 275°C (527°F). The design goal was to obtain 100 hours of continuous operation at temperature.

Test Parameters

Fixed Variables

Speed - 3000 RPM

Evaporator Pressure - 25 psia (240°F)

Figure 3-20. Test Schematic

Figure 3-21. Test Assembly General Arrangement

External temperatures

- Compressor oil bath (527°F) (Monsanto Therminol 66)
- Evaporator oil bath 250°F
- Regulator/expansion valve 250°F
- Expansion valve inlet tube 420°F.

Dependent Variables Range

- Drive torque - 0-4 in. lb
- Suction pressure - 0 to 10 psig
- Discharge pressure - 0 to 1000 psig
- Interstage pressures - 0 to 300 psig
- Interstage temperature - room temp to 600 F.

Test Operations - The actual tests were preceded by component tests and room temperature flow calibrations. The primary component tests involved valving and seals. Room temperature flow calibration involved operating the machine with air or dry nitrogen and measuring flow and torque.

When satisfactory results were obtained from the room temperature flow tests, the system was assembled, leak checked and inserted into the oil bath test chamber. The hot test commenced with gradual heating of the test chamber over a two to three hour period. The chamber heaters were individually controlled with Variacs. Thermocouples sensing chamber wall temperature and internal oil temperature were used to adjust the Variacs.

The compressor was started after the chamber temperature reached 420 to 430°F. The start-up procedure was accomplished by carefully modulating external compressor control valves so as not to load the drive system too rapidly. In later tests, we found that the machine was not particularly sensitive to start-up procedure and could almost be "turned on" at will. We also found that short duration (a few minutes) shutdowns could be tolerated without excessive start-up torque.

3.3.2 Instrumentation

Conventional analog instrumentation components and concepts were used for the test instrumentation system. Temperatures, pressures, speed, torque and weight flow were measured. Temperatures were measured using chromel-alumel (Type K) thermocouples read out on a standard multi-point recorder. Pressures were measured with calibrated strain gage type pressure transducers. Torque and weight flow were measured using strain gaged bending beams as force transducers. Speed was measured with a proximity sensor picking up signals from a 19 tooth sprocket. A block diagram of the system is shown on the next page (Figure 3-23). A discussion of the transducer characteristics and installation parameters is given below.

3.3.2.1 Temperature

Temperatures of 16 fluid streams, compressor block, internal and external heater surfaces were measured with chromel-alumel thermocouples. Thermocouples were spot welded to the surfaces. Internal fluid (inside the 1/8 inch tubes) temperature were measured with thermocouple probes which could be conveniently housed in 1/8 inch OD tube fitting. A typical installation is shown sketched below (Figure 3-22).

Figure 3-22. Temperature Instrumentation

Figure 3-23. Instrumentation Block Diagram

Because the exposed terminals could be at a temperature different than the 16 primary junctions, a separate test was run to determine if this could be an error source. This test, which included exposing the junction and the terminals to temperature differences as high as 400°F, showed that there was no measurable error.

A 24 point thermocouple recorder (Esterline Model E1124E) was used. The chart paper resolution was of the order of $\pm 1^\circ\text{F}$. The following is a list of the thermocouple locations typically used during any test.

1. Stage 1 Suction - Approximately 12" below compressor
2. Stage 1 Discharge - Approximately 12" below compressor
3. Stage 2 Discharge - Approximately 12" below compressor
4. Stage 3 Discharge - Approximately 12" below compressor
5. Compressor Head - Near Stage 3
6. Compressor Body - Near Stage 1
7. Oil Sump - Bottom
8. Shaft Seal - upper end
9. Condensor Tubing - O.D. middle coil
10. Receiver Surface
11. Oil Bath - Near receiver
12. Oil Bath Tank Outside Surface - Near top
13. Oil Bath Tank Outside Surface - Middle
14. Oil Bath Tank Outside Surface - Bottom
15. Pressure Regulator Body
16. Regulator Inlet Fluid
17. Evaporator Oil Bath
18. Evaporator Exit Fluid

3.3.2.2 Pressure

Pressures were monitored both by strain gage transducers and ordinary Bourdon tube gages. Pressures were transmitted to the transducers via 1/8 inch OD tube fittings tee'd into the appropriate sense points. The following pressures were normally monitored:

1. Stage 1 suction
2. Stage 1 discharge

3. Stage 2 discharge
4. Stage 3 discharge
5. Oil pump discharge
6. Regulator inlet
7. Regulator outlet
8. Evaporator outlet

The Bourdon tube gauges were used primarily for quick look monitoring and therefore were not formally calibrated. The strain gage pressure transducers were calibrated end-to-end against a 12 inch diameter Wallace and Tiernan test gage. A shunt calibration resistor selected on the basis of this calibration was inserted into the signal conditioning for each pressure transducer to permit electrical verifications before and after each test sequence.

3.3.2.3 Input Power

The drive motor was instrumented to monitor torque and speed (Figure 3-24). Torque was monitored by measuring the reaction of the motor torque against the gimbal system. The strain gage reaction beam is scaled to yield an output of 0.206 volts per inch-lb. of torque. Typical full scale readings ranged from 2.5 to 3.5 inch-lb. The torque meter was calibrated by hanging weights on the ends of the torque beam.

Figure 3-24. Drive Motor Instrumentation

Speed was measured by a magnetic proximity device (Airpax 14-0001) picking off signals from a 19 tooth x 3 in. diameter sprocket. The Airpax unit contained circuitry to convert pulses to a DC voltage proportional to speed.

3.3.2.4 Flow Metering

No simple, reliable method to measure flow rate on a direct basis was found. Accordingly, gas and water flow measurements were made indirectly by noting the rate of pressure rise in a known volume or by noting the weight change of a liquid storage tank.

Compressor throughput (flow and pressure) is a key performance parameter indicating the "health" of the machine. Room temperature checks of the throughput were measured by recording the pressure vs. time required to fill a bottle of known volume. The flow is directly proportional to the slope of the pressure-time plot. Our curves were produced on an X-Y plotter with a pressure scale of 1 inch per 200 psi and time scaled at 20 seconds per inch. The storage bottle volume was 14.1 cu inches plus an allowance for inter-connecting tubing, typically 1 to 3 cu inches depending on the test set up. Pressure was measured by a calibrated strain gage type transducer scaled to produce 1.0 volt per 1000 psi. Volumetric flow (V, cu-in/min) was computed as follows:

$$\dot{V} = V \times \dot{P}/P_0$$

when V = total volume of gas in container and tubing downstream of discharge, cu-in

P = slope of pressure vs. time psi/min

P₀ = compressor inlet pressure psia

Water flow during the "hot" tests was obtained by measuring the weight change of a pressurized water bottle supplying the compressor suction. This weight measuring bottle was external to the main system in the heated oil and could be valved in or out of the flow loop as required. When this auxilliary receiver was used, the discharge from the primary receiver in the heated oil bath was shut off. The condensate was allowed to collect in the internal receiver.

Since the flow rates were of the order of .01 lb./min. and the flow measuring period was usually less than five minutes, ample excess volume was available without flooding.

The external weight tank was supported on a strain gaged aluminum flexure. The strain gage signal conditioning system was scaled to produce 1.0 volt output per 1.0 lb. change in weight. An external weight was used to calibrate the weight system end-to end.

3.3.3 Results

All-up system tests were conducted in May, June, and September through November. A total of eight hot tests were run. Additional tests were attempted but were aborted before significant data was collected.

The hot tests are summarized in Table 3-1 and in the test log from the runs made with the Part B compressor.

Each of the hot runs was preceded by room temperature calibrations to verify the health of the machine. Results of these tests show the following trends.

- a. Volumetric efficiency ranges from 30 to 75 percent, depending on the condition of the machine. For example, a 30 percent reading means that something is definitely not right. Values between 60 and 75 percent are considered satisfactory.
- b. Volumetric efficiency is essentially constant with speed, discharge pressure and suction pressure.
- c. Drive torque increases slowly with increasing discharge pressure in a "healthy" machine. For example, in a run prior to the September 19 hot test, the torque rise going from 200 psi to 800 psi was only ten percent (3.0 in-lb to 3.3 in-lb).

- d. Effective compressor flow can be obtained only by pressurizing the crankcase by stage 1 discharge pressure. When the crankcase is operated at stage 1 suction pressure, flow rate falls linearly with discharge pressure.
- e. Piston blowby, measured statically does not seem to affect performance significantly. However, based on anomolous results at temperature, blowby may be a significant factor when operating hot.

Table 3-1. Hot Test Summary (1 of 2)

Date	Configuration	Temp. Range	Test Duration	Remarks
May 10, '79	All up system - Basic 1977 compressor with larger oil sump, reworded valves.	450-470°F	1 hour	System check out only.
May 11, '79	Same as May 10 configuration.	420-500°F	3 hours	Shutdown due to high torque - probably due to loss of stage 3 discharge valve. o Lost all static flange seals as well as shaft seal.
June 10,	Same as May 10 configuration. Test set up arranged to measure steam flow rate.	450°F	½ hour	Shutdown due to broken coupling.
June 17	Same as June 10.	440°F-480°F	1½ hours	o Measured flow at .0075 lb/min. Condensing pressure was 625 psi. o Test shutdown due to overheated power supply.
Sept. 19	o All up system. o New compressor block o O-ring discharge valve seats o Graphite-metal seal. o 60u-filter installed at oil pump discharge.	400-527°F.	6½ hours	o Very steady operation until stage 3 valve failure. o Flow measured at .008 lb/min. o Shutdown due to stage 3 valve failure.
Oct. 12	Same as Sept. 19 test except O-ring valve seal replaced with coined all metal seat. Shaft seal bellows brazed to shaft, was previously expoxied.	430-530°F	3 hours	o Shaft seal bellows fracture. o Shutdown from high torque. o Oil filter found plugged. o Interstage pressure oscillation noted.

Table 3-1. Hot Test Summary (2 of 2)

Date	Configuration	Temp. Range	Test Duration	Remarks
Nov. 1	Same as Oct. 12 test except o Plumbing rearranged so that crankcase operates at stage 1 suction pressure. o Brass shaft seal bellows replaced with stainless steel unit.	430-515°F	1½ hours	o Compressor flow less than backflow leakage, resulting in high torque. o Test shutdown when power supply fuse blew.
Nov. 15	Plumbing changed back to Oct. 12 (& previous) configuration.	430-450°F	1 hour	o Significant flow and pressure oscillations noted. o System shutdown after internal plumbing leak and shaft seal failure.

3.3.4 Test Log

3.3.4.1 September 19 Test

CONFIGURATION:

Valving: Size 003 Viton O-rings used for discharge ball check seats. Suction valves made from spot welded .004 inch thick hand formed stainless steel reeds.

Running Gear: (Pistons, cranks, etc.) Same as 1977 machine.

Shaft Seal: Metal-graphite running surfaces. Metal surface held in place with brass bellows bonded to shaft with high temperature epoxy.

Plumbing: Per test schematic - designed to simulate complete refrigeration system. Stage 1 discharge plumbed into oil sump.

TEST OPS SUMMARY:

0930 Heaters turned on

1106 Compressor turned on 3000 RPM
Oil bath temp 400°F

1200 Oil bath temp 500°F
Condenser Pressure 750 psig

1344 Oil Bath temp 527°F (275°C)
Condensing pressure 1000 psig
Torque - 2.7 in-lb
Flow .008 lb/min

1407 Oil bath temp 524°F
Condensing pressure 960 psig
Torque 3.0 in-lb

1513 Oil - 524°F

Condenser 960 psig

Stage 2 pressure oscillations torque noise noted.

Heaters cut back

1621 Oil - 500°F

Condenser - 760 psig

Torque 2.4 in-lb.

Stage 2 pressure and Torque still noisy

1706 Oil - 435°F

Condenser 470 psig

Lost stage 3 discharge valve sealing -

(Stage 2 pressure began to approach stage 3 pressure) compressor shut down.

POST TEST DIAGNOSIS:

1. Stage 3 discharge valves seat failed (ball bottomed out, squeezing O-ring aside). Stages 1 and 2 appear undamaged and still maintain sealing.
2. Shaft seal bellows-to-shaft epoxy bond failed. The graphite-to-metal sealing surface appeared to be in good condition with no visible wear.
3. Running gear, pistons, bearing, crank, etc., show no signs of abnormal wear.

CONCLUSIONS:

1. "Soft" O-ring seals are not adequate for stage 3 conditions.
2. Epoxy is not a satisfactory bonding material for the seal bellows.

3.3.4.2 October 12 Test

CONFIGURATION:

Valving - Same as September 19 except discharge valves are now all metal. The Viton O-ring "soft seat" was replaced by a conical-coined seat.

Shaft Seal - The brass bellows used to provide misalignment correction in the shaft seal was brazed to provide a more reliable joint. The September 19 test had an epoxy bonded bellows.

Pre-test Performance - Room temperature volumetric efficiency ran 60-70 percent; the same as before the previous tests.

TEST OPERATIONS:

1000 - Heaters turned on

1415 - Compressor turned on (3000 rpm)
Oil bath temperature 430°F

1530 - Oil bath temperature
Condenser pressure 555 psig
Torque - 1 - 1 1/2 in-lb (this is low)
Considerable (mechanical) noise from drive system. Stage 1 and 2 pressure oscillations noted - amplitude: $\pm 20\%$ of static discharge pressure
Period: 30 to 60 seconds
Torque variations track stage 2 discharge pressure variations
Oil pressure low (5 psig)

1700 - Oil bath temperature 525°F

Condenser pressure 963 psig

Torque approximately 2 in-lb

Stage 1 and 2 discharge pressure oscillations continuing

Occasionally torque spikes seen without corresponding pressure spikes

1730 - Oil bath 526 °F

Condenser pressure 980 psig

Sudden torque causes system shutdown. Torque rise was not accompanied by pressure changes.

POST TEST DIAGNOSIS:

1. Oil filter found totally plugged. No visible debris. Filter is a Nupro 1/4 tube size X 60 micron sintered bronze unit. Nominal capacity is 4.8 gpm at 100 psi.
2. No visible damage seen on internal running gear (bearings, pistons, etc.).
3. Post test room temperature performance shows slightly lower than nominal performance (60 percent versus 65 percent volumetric efficiency).
4. Characteristic lower stage discharge pressure oscillations still present. These oscillations do not exist when each stage is operated separately.

CONCLUSIONS:

Test shutdown caused by plugged oil filter. Two effects are possible for plugged filter:

1. Loss of lubricant which would lead to increase in friction and hence torque.

2. Oil pump is a positive displacement unit made from a pair of gears. A sudden filter blockage could back up oil pressure (approximately 1000 psig) and cause torque rise. (Oil pump gear is direct connected to crank shaft.)

The sudden torque rise leads us to believe effect #2 above.

3.3.4.3 November 1 Test

CONFIGURATION:

Valving - same as October 12 test

Shaft Seal - Brass bellows replaced with stainless steel unit from Miniflex Corporation, Van Nuys, CA. (Cat. #SS320-30-74. 5/16 ID X 1/2 OD X 7 convolution). This unit appeared somewhat stiffer than the 10 convolution brass unit it replaced. Static and dynamic tests, however, show that it seals quite effectively ("bubble tight").

Plumbing - A major change was made in the plumbing to eliminate the lower stage pressure oscillations. Normally the evaporator return is plumbed directly into stage 1 suction. Stage 1 discharge is then plumbed into the oil sump. In this instance, the evaporator return was plumbed into the oil sump with stage 1 suction drawing out of the sump. This configuration did eliminate the pressure oscillations and did not appear to affect performance, at least at lower (400 psi) discharge pressures.

TEST OPERATIONS

0910 - Started heaters

1045 - Fired up compressor (3000 rpm)

Oil bath 430°F

Condensor pressure 380 psi

Torque 1 1/2 - 1 3/4 in-lb

No Drive system noise as in October 12 test.

1130 - Oil bath temperature 500°F

Condenser pressure 740 psig

Torque 2 in-lb

Can not control suction pressure from slowly rising. This is an indication of either excessive internal leakage or a gross reduction in compressor flow capability.

1200 - Oil bath temperature 515°F

Condensor pressure 880 psig

Compressor shutdown.

Drive system power supply fuse blown from high motor torque. No corresponding pressure rise seen before shutdown.

POST TEST DIAGNOSIS:

1. Disassembly and inspection of internal running gear (bearings, pistons, etc.) show no visible signs of change.
2. Piston ring static blow-by measurements made. Results indicate leakage comparable to pre-September 19 test.
3. With crankcase sump operated at Stage 1, suction pressure tests show that flow rate decreases linearly with discharge pressure. Flow is nearly zero at design discharge pressure of 900 psi.
4. When crankcase/sump is operated at Stage 1, discharge pressure flow rate is constant with increasing discharge pressure.

CONCLUSIONS:

1. Flow can not be produced in sufficient quantity at high pressures when crankcase/sump is operated at stage 1 suction pressure.
2. Lack of flow probably reduced lubricant carry-over to upper cylinder zone resulting in high torque which ultimately caused shutdown.

3.3.4.4 November 15 Test

CONFIGURATION:

Valving - No design changes; but Stage 2 and 3 suction valve reed restraining straps which appeared cracked were replaced.

Shaft Seal - No change

Plumbing - Changed to October 12 configuration, that is, evaporant discharge fed directly into stage 1 suction. Crankcase/oil sump pressurized at stage 1 discharge level.

TEST OBJECTIVE:

The objective of this final test was to determine if a stable long duration operation could be obtained at a lower temperature. Our plan was to first stabilize at 450°F and then gradually increase the temperature until anomalies developed.

TEST OPERATIONS:

0900 - November 15 - heaters turned on

1145 - Started compressor (3000 rpm)

Oil bath temperature approximately 430°F

Condensor pressure approximately 350 psi

Drive noise is quite pronounced, similar to October 12 test. Shaft seal leak detected after about 15 minutes operation.

1200 - Oil bath temperature approximately 450°F

Condensor pressure approximately 400 psig

Torque oscillations and drive noise quite pronounced

Stage 1 discharge pressure falling steadily

1213-1245

Lost discharge pressure, all attempts to get system back up failed - test shutdown when oil pressure lost.

POST TEST DIAGNOSIS:

1. Leak found in tubing fitting at stage 3 discharge. This fitting, an 1/8 inch brass Swage Lock unit, had apparently just worn out from repeated use.
2. Shaft seal bellows found fractured. Crankcase pressure was held by back-up elastomer Balseal. However, slow leak developed in the oil scavenge tube placed between main metal-graphite seal and Balseal and caused loss of lubricant supply.
3. Room temperature flow and leak checks indicated that all valves except stage 1 suction to be in good working order. All (statically measured) leakage rates appear to be less than 5 percent of nominal flow.
4. During final checkout, Stage 1 suction valve appeared to "snap" into place with nominal performance once again. Inspection of the valve assembly shows some evidence of debris but no physical distortion could be seen.

CONCLUSIONS:

Test shutdown resulted from a loss of refrigerant (water) through a leak in a high pressure side fitting. The shaft seal and stage 1 valve failures would also have caused eventual shutdown had the fitting not leaked.

The flow and torque oscillations noted early in the test may have contributed to the failure of the shaft seal.

3.3.4.5 Test Data

Data taken from the September 19, October 12, and November 1, tests are shown on the following six pages.

Table 3-2. Hot Test Data (1 of 6)

ALL-UP SYSTEM TEST

SPEED: 3000 RPM

DATE: 19 SEPT. 1978

TIME OF DAY	TORQUE (IN-LB)	D ₁ (psig)	D ₂ (psig)	D ₃ (psig)	S ₁	T _{OIL} (°F)	T _{COND} (°F)	T _{EVAP} (°F)
1106*	0.00		0	0	0.0	405		230
1113**	2.20		100	320	3.0	410		232
1120	2.30		115	365	5.0	428		232
1140	2.10		105	520	1.0	465		230
1150	2.10		98	600	0.0	476		225
1204	2.35		117	750	1.2	500		228
1217	2.15			785	-0.5	509		225
1225	2.35			815	1.1	510		225
1235	2.35			850	0.5	514		222
1244	2.40			890	1.0	518		225
1253	2.53			910	1.2	520		230
1302	2.55			940	1.2	524		236
1310	2.55			950	0.5	525		236
1319	2.60			965	1.2	525		235
1328	2.62			970	0.5	525		230
1344	2.70			1000	1.2	527		225
1352	2.90			990	1.0	527		225
1407	3.00			960	1.0	524		223
1445	2.90			930	1.0	520		220
1504	3.00			950	1.0	523		220
1513	3.20		220	960		524		
1523	3.15		220	950		523		219
1532	3.30		240	940		519		244
1545	2.70		190	900		516		240
1602	2.80		205	880		514		
1621	2.40		170	760		500		
1639	2.20		155	600		482		
1648	2.40		175	570		465		
1657	2.30		164	515		454		
1706	2.25		160	470		435		

* START

** SPEED = 2000 RPM

Table 3-2. Hot Test Data (2 of 6)

ALL-UP SYSTEM TEST

SPEED: 3000 RPM

DATE: 12 OCT. 1978

TIME OF DAY	TORQUE (IN-LB)	D ₁ (psig)	D ₂ (psig)	D ₃ (psig)	S ₁	T _{OIL} (°F)	T _{COND} (°F)	T _{EVAP} (°F)
1415*	1.2			START	8	430		235
1429	1.5	30	70	400	8	444		235
1447	1.5		65	490	8	460		240
1502	1.4		65	550		475		241
1517	1.5		75	635		488		243
1529	1.8		90	720		499		244
1539	1.6		80	790		509		245
1556	2.2	35	110	830		513		242
1609	1.9	30	80	850		514		242
1616	2.2	35	100	870		515		240
1626	1.9	25	90	890		517		240
1640	2.1	30	100	920		521		242
1649	2.0	35	100	945		523		242
1701	1.8	20	90	963		525		245
1708	1.9	30	100	970		525		240
1718	1.9	25	100	975		526		243
1730**	HIGH	25	45	980		526		243

* START

** SHUT DOWN

Table 3-2. Hot Test Data (3 of 6)

ALL-UP SYSTEM TEST

SPEED: 3000. RPM $\pm$ 5%

DATE: 1 NOV. 1978

TIME OF DAY	TORQUE (IN-LB)	D ₁ (psig)	D ₂ (psig)	D ₃ (psig)	S ₁	T _{OIL} (°F)	T _{COND} (°F)	T _{EVAP} (°F)
1049		30	70	400	10	444		
1055	1.56	27	65	440	7	460		
1105	1.70	27	65	550	7	470		
1125	1.95	40	90	700	10	490		
1131	1.85	40	85	740	10	500		
1144		40	100	800	10	506		
1201	2.20	50	120	850	7	515		
1210	2.50	60	140	860	10	515		
1220		65	130	880		515		
1223*	HIGH							

*SHUT DOWN

CONFIGURATION:

- DELETE OIL FILTER
- NEW S/S BELLOWS
- NO EVAPORATOR
- NEW PLUMBING SUMP
(STAGE 1 SUCTION)

Table 3-2. Hot Test Data (4 of 6)

3-55

Table 3-2. Hot Test Data (5 of 6)

Table 3-2. Hot Test Data (6 of 6)

4.0 CONTROLS DEVELOPMENT

A downhole vapor cycle refrigeration system requires two basic control elements. The first is an expansion valve which controls evaporator vapor pressure, and second, a means to permit safe downhole start-ups.

The first requirement can be met by a modified gas pressure regulator. Tests with a Consolidated Controls #2338 gas regulator showed that it could be used to control the expansion of high pressure (1000 psi) high temperature (400°F) water at rates in the .01 lb/min flow range. The required modifications include substitution of an all-metal seat and the replacement of existing seals with Viton O-rings. Further structural changes shown on the accompanying drawing are made to fit the unit inside a 2-1/4 ID Dewar.

A model of this valve was used in all of our system tests without noticeable malfunctions. The May-June tests were run with an unmodified valve having Kel-F seats. The later September-November tests were run with all-metal seats. It should be noted, that for use downhole, the valve body itself must be located in a thermally protected area--not because of its inherent temperature limitations, but for reasons of heat loss. The valve body, if exposed to the downhole thermal environment, would absorb so much heat that no useful cooling could be accomplished by the expanded water vapor.

The second requirement, the downhole startup, is not a problem for short duration (minutes) shutdowns. A copy of a test plot, Figure 4-1, shows that drive torque for startup is only marginally higher than running torque.

Figure 4-1. Typical Start-up Torque

NOTES:

1. COMPONENT SAME AS USED IN REGULATOR P/N 2341-1 OR MODIFIED AS NOTED.

2. BASED ON CONSOLIDATED CONTROLS CORP. PRESSURE REGULATOR P/N 2341-1 (METAL DIAPHRAGM & SEAT)

LAYOUT-PROPOSED PRESSURE REGULATOR FOR DOWNHOLE REFRIG.
 CH. 5/4/70 SCALE: 2x SIZE

Figure 4-2. Modified Pressure Regulator

1. The first part of the report deals with the general situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the social and economic conditions of the country.

2. The second part of the report deals with the political situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the political conditions of the country.

3. The third part of the report deals with the economic situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the economic conditions of the country.

4. The fourth part of the report deals with the cultural situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the cultural conditions of the country.

5. The fifth part of the report deals with the social situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the social conditions of the country.

6. The sixth part of the report deals with the legal situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the legal conditions of the country.

7. The seventh part of the report deals with the educational situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the educational conditions of the country.

8. The eighth part of the report deals with the health situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the health conditions of the country.

9. The ninth part of the report deals with the housing situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the housing conditions of the country.

10. The tenth part of the report deals with the transportation situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the transportation conditions of the country.

11. The eleventh part of the report deals with the communication situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the communication conditions of the country.

12. The twelfth part of the report deals with the energy situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the energy conditions of the country.

13. The thirteenth part of the report deals with the environment situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the environment conditions of the country.

14. The fourteenth part of the report deals with the security situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the security conditions of the country.

15. The fifteenth part of the report deals with the foreign relations situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the foreign relations conditions of the country.

16. The sixteenth part of the report deals with the internal security situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the internal security conditions of the country.

17. The seventeenth part of the report deals with the defense situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the defense conditions of the country.

18. The eighteenth part of the report deals with the international relations situation of the country and the position of the various groups of the population. It is a very interesting and informative study of the international relations conditions of the country.

For longer duration shutdowns this would not be true, particularly with existing all-metal seat valves. Neither the present Stage 3 ball check nor the Consolidated Controls regulator will shut off bubble tight. To permit startup, one of two things must be done:

1. Isolate compressor from balance of system with zero leak valves
2. Provide actuated bypass around compressor allowing discharge flow back to suction.

The problem of longer duration was attacked on three fronts:

1. Vendor survey of solenoid valves
2. Design of rotary solenoid actuated manual valve
3. Development of leak tight soft valve seats.

Vendor Survey - A survey of likely vendors of high temperature solenoid valves proved negative. For example, two likely vendors, General Controls, Burbank, California, and Circle Seal, Anaheim, California, were interested from a technical standpoint but flatly refused to get involved in a development program. They were also very reluctant to promise leak tight shut-off particularly at temperature.

Rotary Solenoid Actuation - Existing hardware can be adapted to make shutoff or bypass possible. High temperature hand valves and high temperature solenoids are available. They can be combined in a single package to control shut-off remotely. The drawing, Figure 4-3, illustrates one attempt.

The valve is a Nupro-T4W high temperature unit. Bench tests indicate that it will shut-off with 7 in-lb of torque at room temperature. Nupro data indicates shut-off torques up to 20 in-lb at temperature. The valve driver is a pair of Cliftronic rotary solenoids. The assembly, 2-3/8 Diameter X 12 inch long, is designed to fit in the space under the condenser coils. The Cliftronic Solenoids require 28VDC X 9 amps for full closure.

No further work was done on this concept.

Soft Seat Development - The high temperature properties of Kalrez suggested a solution to the leak tight seal problem. A spring load ball check was designed and fabricated around a Kalrez O-ring with a .140 Diameter ball. A photo of the test valve is shown below.

Figure 4-4. Photo - Soft Seat O-ring Check Valve

The valve was leak tight from ambient to 550°F. In fact, it held leak tight over a weekend after cool down to ambient. This concept is attractive since it is easier to implement than metal seats. No special care is required in the finish of the O-ring grooves. In fact, a standard 32 -in machine finish is satisfactory. The major unknown about this soft seat concept is the life after repeated cycling. Similarly configured valves failed after 3 to 4 hours in stage 3. They did not however fail in either Stage 1 or Stage 2. This suggests that long life can be expected if forces can be kept low.

Figure 4-3. Rotary Solenoid Driver Shut-Off Valve

5.0 MOTOR PROCUREMENT

This task was originally to include the development of a procurement specification, vendor survey and selection, motor procurement, test and the procurement of a power supply suitable for use in the field. However, because of delays in the compressor development, this task was essentially "put on hold" after the bids were received. The discussion that follows describes the motor specification and vendor selection.

5.1 MOTOR SPECIFICATION

The motor design requirements are driven by the following considerations:

- a. Environment
- b. Output power and speed
- c. Basic electrical type, e.g., AC or DC? If AC, what frequency, single phase or three phase? If DC, shunt wound or series?
- d. Packaging and envelope

Ideally the design details should be established after the compressor operating parameters are well established. However, because the motor was considered a long lead item, the specifications were finalized early in the program based largely upon data from Phase I. A copy of the specification is shown in Appendix II. A brief summary of the specification is given below.

The motor environment was set at:

Temperature - 275°C

Pressure - 7500 psig

The output power and speed were set at:

Power - 200 watts output

Speed - 3000 RPM

Torque - 5.6 in-lb (based on 200 watts mechanical power output)

The input power was to be:

3 phase, 60 cycle AC

Voltage - TBD

A three-phase induction motor was selected because it has the advantage of requiring no starting controls, brushes or local switch closures.

Furthermore, since the electrical windings are in the non-rotating stator, cooling is much easier to accomplish.

Standard 60-cycle power was selected over the more versatile 400-cycle power because the motor would be used in remote sites. In the case of 60-cycle power on site, it may be only single phase and probably not the desired voltage. Accordingly the plan was to use a commercially available three-phase 60-cycle field generator producing approximately 440 vac. A three-phase Variac would then reduce this so the desired motor terminal voltage could be obtained.

To ease the motor design problems, it was decided not to constrain the motor terminal voltage to any set value. This implied the use of a downhole transformer since practical motor voltages were in the range of 50 to 150 volts. Higher voltages are difficult to achieve because of insulation breakdown at high downhole temperatures.

The shell OD and the requirement for instrumentation cable passage constrained packaging. To minimize the motor structural design requirements, a separate motor pressure hull was specified (see Diagram TNT-41-08). It was to be made from 17-4PH heat-treated to 150 ksi. The motor pressure shell was to be 3.50 OD. The pressure shell ID was set at 2.75 in. Space for instrumentation cabling was formed by .15 deep x .15 wide grooves cut inside the pressure hull.

Another packaging consideration was the requirement for a gear box. Since the compressor crankshaft is offset from the centerline by 0.700 in., a direct drive motor is not feasible. In the original specification the gear box was located on the motor. In retrospect, this may not have been the best choice. Subsequent design studies show that the preferred location of the gear box is

on the compressor. The main advantage of locating the gear box on the compressor is that it can be integrated with the shaft seal and lubrication system. The present design layout shows this preferred arrangement. The motor bids, however, were made on the basis of an integral gear box.

5.2 VENDOR SELECTION

Bids were solicited from two vendors:

American Electronics - \$ 34K
Fullerton, California

Skurka Engineering Company - \$60K
Los Angeles, California

Inquiries to several other motor manufacturers were made but no interest was shown. The primary reason was insufficient production potential.

American Electronic's bid is for a motor sized to produce 200 watts (mechanical equivalent) power output with a geared output shaft speed of 3000 RMP. It is a star connected 3 , 60 cycle AC motor, with 115 volts line-to-line. Current draw is estimated to be 2.8 amperes per phase. The motor is packaged in a 3.5 in. OD pressure hull 10.5 in. long. The geared shaft is offset 0.700 in. from the centerline. The stator is sized to take either the ceramic fiber insulated Gulton wire or the ceramic coated Cooner Sales wire. Delivery is 24 weeks ARO.

Skurka's motor is not described in their bid but from earlier contacts it was designed to use a high temperature enamel rather than ceramic insulation. Skurka seemed to have second thoughts about the design of the gear box under high temperature conditions. Their high bid reflects this. Delivery is quoted as 30 weeks.

6.0 SYSTEM DESIGN AND INTEGRATION

The objective of this task was to design and fabricate a complete refrigerator system suitable for use in a downhole sonde. However, because of a lack of progress in the compressor, only the system design was performed. A discussion of the design follows. Fold-out drawings of the compressor and refrigerator assemblies follow.

6.1 GENERAL PACKAGING STRATEGY

The general arrangement of the required refrigerator components is shown on Figure 6-1. The components from the top are:

1. Cable Head
2. Motor/Gear Box
3. Compressor
4. Oil Sump/Pump
5. Intercooler/Condenser
6. Controls
7. Payload

The components are attached to each other by means of bolted flanges. The flanges are made up with 12 4-40 socket head cap screws. Except for holding internal fluid pressure, these flanges do not have to be leak tight to external brine pressures because sealing is to be accomplished by a 1/16 inch thick Inconel Shell brazed to the top and bottom of the sonde. This shell, designed to be expendable, would be attached after assembly and checkout. The net diameter allowed for the components would be 3.50 inches. Except for the payload end and possibly the cable head, no seals are required to withstand high pressure hot brine. The structural requirements to withstand 7500 psi are modest. For example, a pressure vessel made from 17-4PH heat treated to 150 ksi requires a wall thickness of less than 1/10 inch. The actual thickness would be higher to account for flanges, cutouts, etc. Drawing TNT-2481 shows thicknesses varying from 1/4 to 5/16 inches.

The details of each of the main components is described further below.

SDC SYSTEM DEVELOPMENT CORPORATION	
FOR ANALYSIS AND DESIGN	
GENERAL ARRANGEMENT--COMPRESSOR	
DYNAMICS SUPERCOMPUTER, PART II	
DATE: 14188	REV: TNT-2480
6-30-18	1 OF 1

Figure 6-1. General Arrangement--Compressor

6.2 MOTOR/GEAR BOX

The motor is housed in a flanged pressure shell, 3.50 O.D. The upper end mates with an adapter to be used in case of a system test in a high pressure auto-clave. The shaft end of the motor bolts to a shaft seal/gear box assembly. This gear box provides a small (3000/3450) speed reduction as well as providing alignment for the offset crank. The shaft seal is similar to the unit tested except it has two sealing surfaces in a series. Shaft and gear bearings are standard .2500 X .6250 radial inch series. The gear design has not been finalized but would be made from heat treated gear stock. The gear box is supplied with lubricant from the oil pump through a passage drilled in the block.

Shaft connection is made with a broached hex coupling which is inherently trapped in place. This permits assembly and disassembly simply by a push fit. No keys or lock nuts are required. A broached coupling was used during the system test after repeated failures with couplings that were keyed.

6.3 COMPRESSOR/OIL SUMP

The compressor shown on the assembly uses the same piston-crank assembly as the Phase I machine. The block has the same proportions as the Part B Phase II machine. The valving is the same as used in the last systems tests.

The oil sump/pump assembly bolts directly onto the compressor block. Tubing feed-through seals as developed in the Phase II component development program are used. The oil pump, identical, to the original, is located at the bottom of the sump. The inlet is screened (325 mesh). No discharge filter is used.

Tube fittings (1/8 Swagelock) are welded to the bottom of the sump. These fittings form the mechanical interface with the heat exchanger section of the refrigerator.

The lubricant for the compressor is a perfluorinated poly-ether (Brayco 815Z) made by the Bray Oil Company, Los Angeles. Based upon qualitative observation made during the high temperature tests this oil appears to be reasonably compatible with both water (vapor) and compressor internals. No visible signs of breakdowns or chemical attacks were seen on surfaces exposed at temperature to Brayco.

Bravco does not appear to be miscible with water at conditions approximating the liquid receiver (900 psi x 275°C). Furthermore, since it has a specific gravity nearly twice that of water it will sink to the bottom of the receiver. This permits it to be recovered and recirculated back to the oil reservoir. Other properties of this oil are on the specification sheet shown reproduced on the following page.

BRAY OIL COMPANY

Hughes
HMS 20-1830

BRAYCO® MICRONIC 815Z HYDRAULIC FLUID AND LUBRICATING OIL, SYNTHETIC, ROCKET PROPELLENT COMPATIBLE, HIGH TEMPERATURE

DESCRIPTION: BRAYCO® 815Z is a clear water white fluid. The fluid's base is a perfluorinated poly-ether. The fluid is practically chemically inert, compatible with rocket propellents and oxidizers, is unaffected by ultra violet, cosmic radiation, or high vacuums. It has an exceptionally high viscosity index and low volatility, and has little tendency to form deposits. It exhibits excellent lubricating properties, good dielectric properties, excellent shear stability and low toxicity.

USES: BRAYCO® 815Z is practically insoluble in current fuels and oxidizers, as well as in most organic solvents. BRAYCO® 815Z is intended for aerospace use, and has been used successfully in many space missions. It is particularly suited for deep space missions where extreme vacuums and wide temperature variations, -84.4°C to 260°C (-120°F to 500°F) are routine.

LIMITATIONS: BRAYCO® 815Z may be adversely affected by Lewis Acids catalysts such as $AlCl_3$ at elevated temperatures. Newly exposed rubbing surfaces of aluminum or magnesium at temperatures of 205°C (400°F) and above should be evaluated for corrosiveness.

PACKAGING: BRAYCO® 815Z is packaged in one pound containers. Other containers are available on special request.

PROPERTIES

Method (ASTM)	Test	Typical
D-287	Gravity, Specific 15.5°C/15.5°C (60°F/60°F)	1.866
Table 8	Pounds/gallon	15.4
D-445	Viscosity @ 205°C (400°F), cs	11
D-445	Viscosity @ 98.9°C (210°F), cs	40
D-445	Viscosity @ 37.8°C (100°F), cs	129
D-445	Viscosity @ -17.8°C (0°F), cs	450
D-445	Viscosity @ -40°C (-40°F), cs	2600
D-445	Viscosity @ -53.9°C (-65°F), cs	6700
D-2270	Viscosity Index	350
D-92	Flash (Halogen Halo)	299°C (570°F)
D-97 Mod	Pour Point	< -73°C (-100°F)
HMS 20-1830	Vapor Pressure, Torr 38.8°C (100°F) 205°C (400°F)	< 1×10^{-9} < 1×10^{-6}
NASA SP-R-0022	Vacuum Weight Loss, %	0.8
NASA SP-R-0022	Volatile Condensable Material, %	.05
FTM-3009	Solid Particle Contamination, Number of Particles per 100 ml	
	5-15 microns	150
	16-25 microns	45
	26-50 microns	23
	51-100 microns	3
	+100 microns	Nil

ALT1

BRAY OIL COMPANY 1925 N. MARIANNA AVENUE, LOS ANGELES, CALIFORNIA 90032

2131 575-1212

TWX: 910-321-3540

CABLE: BRAYOIL LOSANGELES

6.4 HEAT EXCHANGER SECTION

The heat exchangers (intercoolers & condenser) are made with 1/8 OD stainless steel tubing spirally wrapped around 3.50 shell with 3/8 walls. The shell is grooved to accept 1/8 OD tubing so that the tubing will be just flush with the surface.

The computed lengths required for each heat exchanger are quite long:

1-2 intercooler 10 ft.

2-3 intercooler 11 ft.

Condenser 132 ft.

The total length of the heat exchanger section is nominally 40 inches, based on a 1/4 pitch tubing wrap. The system tests indicate that far less tubing is required. For example, no performance change was noted when the amount of tubing was cut in half. Additional study is required to define minimum acceptable lengths.

The bottom of the heat exchange system terminates in an integral water reservoir. It is sized to hold a pint of water.

6.5 PAYLOAD/CONTROL

The payload and controls are located inside a 2-1/4 ID Dewar, which in turn is slipped inside a 3.0 ID x 3.5 OD 17.4 PH pressure hull. The control elements are (1) a pressure regulator and (2) a liquid-to-vapor heat exchanger.

The pressure regulator, a cut-down Consolidated Control #2338 Regulator, fits snugly upside-down inside the Dewar. The vapor-liquid intercooler is formed by a pair of 1/8 OD tubes brazed together in a helix 5 turns x 2" OD.

The simulated payload is a 100 watt electrical immersion heater fixed inside a 1-1/2 OD x 5" long copper cylinder. The payload cylinder is wrapped with 20 coils of 1/8 OD tubing brazed in place. This cylinder represents the evaporator cold plate.

Approximately 12 inches of high temperature insulation ($K = 0.032$ or better) closes out both ends of the Dewar. The overall length is approximately three feet. A double gasketed end closure is shown only for illustrative purposes. A real closure would require transducer ports or feed-throughs.

**APPENDIX I
THERMODYNAMIC ANALYSIS**

(TM-(L)-6108/202/00)

**Prepared by Alan Haire
October, 1978**

REFRIGERATION SYSTEM COMPONENTS

Major components of the refrigeration system are shown in the schematic Figure 1 and described below, starting at the compressor inlet.

Compressor Each of the three stages is assumed to raise the pressure of the vapor by $\sqrt[3]{\text{compressor pressure ratio}}$ and raises the temperature by adiabatic compression.

Intercoolers Cool vapor by rejecting heat to the brine.

Check Valve Prevents high pressure vapor between the check valve and the pressure regulator from flowing back to the compressor outlet when the compressor is shut off. The small amount of high pressure vapor between the check valve and the compressor outlet will leak back past the compressor pistons so there is not a large pressure across the compressor. Thus, the torque when the compressor is restarted will be minimized.

Condenser Cools and then condenses the vapor by rejecting heat to the brine.

Liquid Receiver Provides a reservoir of fluid for transients and also to make up for leakage out of the system.

Subcooler Cools the hot liquid by transferring heat to the cool vapor before it returns to the compressor. This is not necessary for operation of the system, but it increases the cooling capacity.

Pressure Regulator Causes constant enthalpy expansion to a lower pressure, reducing the fluid temperature to that required for payload cooling. Also, it controls the compressor inlet pressure.

Payload A nominal 50 watt source of heat that must be maintained at 125°C.

Dewar An insulated compartment that restricts the heat leak into the cool parts of the system.

Figure 1. Refrigeration System Schematic

REFRIGERATION SYSTEM CHARACTERISTICS

Analyses and tests of the system showed that it would have the following characteristics.

Pressure drops due to fluid friction are negligible. Thus, the pressure from the compressor outlet to the pressure regulator inlet can be considered constant, and the pressure from the regulator outlet to the compressor inlet can also be considered constant.

The line from the subcooler outlet to the compressor inlet cannot be insulated sufficiently to keep it cool. The fluid in it is therefore assumed to reach the brine temperatures. This is a conservative assumption.

The compressor inlet pressure (pressure regulator outlet pressure) of approximately 20 psia will provide sufficient flow for 50 watts of gross cooling.

Compressor outlet pressure (condenser pressure) will be the saturation pressure at the condenser temperature.

Cooling in the intercoolers is so efficient that the fluid will almost reach brine temperature.

Complete system thermodynamic performance, including substantiation of some of the above characteristics, were performed. Summaries of the analysis are below.

Heat Transfer Rates Preliminary values of thermodynamic characteristics of the system were used to calculate conduction and convection rates. Conservative values were used throughout; so they were not recomputed for the final thermodynamic cycle. Rates are:

Vapor convection	41.4 BTU/hr ft ² °F
Liquid convection	164
Vapor condensation	350
Brine convection	56.4
Tube conduction	4,442

Thermodynamic Cycle Thermodynamic cycles were analyzed for the following conditions:

	<u>Compressor Inlet Pressure</u>	<u>Ambient Temp.</u>	<u>Volume Flow Rate</u>	<u>Gross Cooling Capacity</u>	(not considering heat leak)
	<u>psia</u>	<u>°C</u>	<u>ft³/min</u>	<u>watts</u>	
1)	30	275	.14	95	(Nominal Design value.)
2)	30	300	.14	84	
3)	30	220	.14	116	
4)	30	150	.14	150	
5)	20	275	.14	64	
6)	15	275	.14	48	
7)	30	275	.11	75	
8)	30	275	.19	126	

Heat Exchange Adequacy of the heat exchangers was checked for the thermodynamic cycles. Heat exchanger capacities, assuming a constant ΔT for all cycles, are:

<u>Heat Exchanger</u>	<u>Length</u>	<u>UA</u>
	<u>ft</u>	<u>BTU/hr °F</u>
1-2 Intercooler	22	5.83
2-3 Intercooler	11	2.92
Condenser	132	93
Subcooler	2.5	.738

Safety factors are:

Cycle	1-2 Intercooler	2-3 Intercooler	Condenser	Subcooler
1	2.2	2.2	1.5	1.5
2	2.3	2.3	1.8	1.5
3	2.1	2.2	1.2	1.7
4	2.3	2.4	.9	1.7
5	3.2	3.3	2.3	1.5
6	4.2	3.3	3.1	1.6
7	2.8	2.8	1.9	1.7
8	1.6	1.6	1.1	1.7

Pressure Drop Preliminary values for the thermodynamic cycle were used for the pressure drop calculations. The drops were negligible; so the analysis was not repeated for the final cycle. Results were:

<u>Heat Exchanger</u>	<u>Pressure Drop-psi</u>
1-2 Intercooler	.5
2-3 Intercooler	.2
Condenser	.3

Dewar Heat Leak Heat leak into the dewar was calculated for 275°C (527°F) ambient brine. The total was 8.3 watts, which is acceptable since the total cooling capacity is 94.6 watts.

Lower Operating Limit If the ambient brine temperature is too low, the fluid reaching the compressor inlet will not be completely vaporized. Liquid could damage the compressor; so this sets the lower operating limit of the system. For 50 watts of refrigeration the lower limit is 260°F with the pressure regulator set at 30 psia. If the regulator is set lower, the lower limit will be lower. The penalty for a lower regulator setting is reduced capacity.

Simulated Payload A simulated payload was designed and analysed. It was sized to provide up to 200 watts of heat.

Compressor Torque A theoretical expression for compressor torque was derived, and constant values in the expression were evaluated to give the following equation:

$$T = 2.0 + .031 P_{in} \left[R^{1/3} - R^{1/4} \right] \left[1 + .360 R^{1/3} + .122 R^{2/3} \right]$$

where T = torque - in lb.

P_{in} = inlet pressure - psia

R = ratio of outlet absolute pressures

Compressor torque and input power for the thermodynamic cycles are:

<u>Cycle</u>	<u>Torque</u> in-lbs	<u>Power</u> watts
1	4.4	150
2	5.3	190
3	3.0	110
4	2.2	80
5	4.3	150
6	4.3	150
7	4.4	150
8	4.4	150

Performance The above tabular data is presented graphically on the pages that follow:

PREPARED A. Hain DATE 12-28-77

SECTION NO.

SHEET 1 OF 1

CHECKED _____ DATE _____

REFERENCES

- 1) MRI - U - 5427-03 "PROPOSAL FOR PHASE II DOWNHOLE REFRIGERATOR DEVELOPMENT", MRI, 7-28-77
- 2) STEAM TABLES "THERMODYNAMIC PROPERTIES OF STEAM", 1ST ED, KEENAN & KEYES, MAY 1949
- 3) MRI - TMT(2941)FL-1 "FINAL REPORT, CONTRACT EY-76-C-03-1309, MECHANICAL REFRIGERATOR FOR DEEP WELL SONOC ELECTRONICS", MRI, 8-12-77
- 4) MARKS "STANDARD HANDBOOK FOR MECHANICAL ENGINEERS", 17TH ED, BAUMGISTER & MARKS.
- 5) ASME B1PV CODE "ASME BOILER AND PRESSURE VESSEL CODE"
- 6) SAE "SAE AERONAUTICAL INFORMATION REPT. NO. 24, AIRPLANE AIR CONDITIONING ENGINEERING DATA, HEAT TRANSFER, 2-1-52
- 7) MCADAMS "HEAT TRANSMISSION" 3RD ED, MCADAMS
- 8) PERRY "CHEMICAL ENGINEERS HANDBOOK", 3RD ED, PERRY.
- 9) DAUSHERTY "HYDRAULICS" 4TH ED, DAUSHERTY

PREPARED G. Hamb DATE _____

SECTION NO. _____

SHEET 2 OF _____

CHECKED _____ DATE _____

30 PSIA / 275°C / .14 FT³/MINTHERMODYNAMIC CYCLECOMPRESSOR INLET $T \approx \text{AMBIENT} = 275^\circ\text{C} = 527^\circ\text{F}$ INTERCOOLER OUTLET $T = \text{AMBIENT} + 1^\circ = 528^\circ\text{F}$ CONDENSER $T = \text{AMBIENT} + 5^\circ = 532^\circ\text{F}$ COMPRESSOR INLET $P = 30 \text{ PSIA}$ COMPRESSOR OUTLET $P = 900 \text{ PSIA}$ (SAT. AT 532°F)COMPRESSOR PRESSURE RATIO $= 900 / 30 = 30$ STAGE PRESSURE RATIO $= \sqrt[3]{30} = 3.11$ COMPRESSOR INLET $\rho = .0513 \text{ LB/FT}^3$ ($30 \text{ PSIA} \text{ \& } 527^\circ\text{F}$)VOLUME FLOW RATE $= .14 \text{ FT}^3/\text{MIN}$ $\dot{W} = (.0513 \times .14) = .00714 \text{ LB/MIN} = .431 \text{ LB/HR}$ FOR FOLLOWING ANALYSIS P IN PSIA, T IN $^\circ\text{F}$, s IN BTU/LBSTAGE 1 IN $P = 30$ $T = 527$ $s = 1.8596$ $h = 1298.7$ STAGE 1 OUT = 1-2 INTERCOOLER IN $P = (30 \times 3.11) = 93$ $s = 1.8596$ $T = 528$ $h = 1438.3$ 1-2 INTERCOOLER OUT = STAGE 2 IN $P = 93$ $T = 528$ $s = 1.7314$ $h = 1293.8$ STAGE 2 OUT = 2-3 INTERCOOLER IN $P = (93 \times 3.11) = 290$ $s = 1.7314$ $T = 528$ $h = 1432.5$ 2-3 INTERCOOLER = STAGE 3 IN $P = 290$ $T = 528$ $s = 1.5916$ $h = 1275.2$ STAGE 3 OUT = CONDENSER IN $P = (290 \times 3.11) = 900$ $s = 1.5916$ $T = 532$ $h = 1406.9$ CONDENSER VAPOR-LIQUID INTERFACE $P = 900$ $T = 532$ $h = 1195.4$ CONDENSER OUTLET = SUBCOOLER HOT SIDE INLET $P = 900$ $T = 532$ $h = 526.6$ SUBCOOLER HOT SIDE OUTLET = REGULATOR INLET $P = 900$ ASSUME $T = 420$ $h = 414.6$

PREPARED C. Howe DATE 9-14-78

SECTION NO.

SHEET 2 OF

CHECKED _____ DATE _____

PERFORMANCE

INLET PRESS = 30 PSIA

VOL FLOW = .14 FT³/MIN

---- INSUFFICIENT CONDENSER AREA

PREPARED A. Haws DATE 9-14-78

SECTION NO.

SHEET 3 OF

CHECKED _____ DATE _____

PERFORMANCE

AMBIENT TEMP.

275° C

VOL. FLOW

.14 FT³/MIN

PREPARED A. Hair DATE 9-14-78
CHECKED _____ DATE _____

SECTION NO.

SHEET 4 OF

PERFORMANCE

AMBIENT TEMP.

275°C

INLET PRESS.

30 (5) A

PREPARED A. Hain DATE 9-14-78

SECTION NO.

SHEET 5 OF

CHECKED _____ DATE _____

PERFORMANCE

INLET PRESS. 30 PSIA
VOL. FLOW. .14 FT³/MIN

PREPARED A. Howe DATE 9-14-78

SECTION NO.

SHEET 6 OF

CHECKED _____ DATE _____

PERFORMANCE

INLET PRESS.

30 PSIA

VOL. FLOW.

.14 FT³/MIN

APPENDIX II

PROCUREMENT SPECIFICATION

HIGH TEMPERATURE GEARED MOTOR

**PROCUREMENT SPECIFICATION
HIGH TEMPERATURE GEARED MOTOR**

This specification describes the procurement and performance requirements for a geared 3 phase, 60 cycle induction motor suitable for use in driving a downhole electronic cooling refrigerator.

SECTION 1 - TECHNICAL REQUIREMENTS

1.1 PERFORMANCE (At Temperature and Referred to Output Shaft)

- o Torque: 5.6 in-lb
- o Speed: 3000 $\pm$ 10% RPM
- o Input Power: 3 phase, 60 cycle voltage and current to be specified, but current is not to exceed 2.5 amperes per phase at rated load
- o Life: 250 hours

1.2 OPERATING ENVIRONMENT

- o External media - Hot corrosive brine
- o Temperature of brine 275°C
- o Brine ambient pressure 7500 psi
- o Motor internals can be subjected to superheated steam at two atmospheres and 275°C which might leak past compressor shaft seal.

Note: Motor to be designed to withstand temperature and pressure environment. Protection from corrosive brine to be provided by a buyer installed Inconel shell.

1.3 ENVELOPE/INTERFACE (See Attached Drawings)

- o O.D. of pressure shell 3.50 inches
- o Length - as required, but not exceed 15 inches
- o Attached interface:
 - 12 - 4-40 screw holes 5/16 deep on 3.250 bolt circle, both ends
- o Drive Shaft:
 - .250 Hex .50 long offset 0.700 from centerline
- o Special cable feed through provisions
 - Provide 8 longitudinal grooves on inner side of motor pressure shell. Grooves to be 9.15 deep by .075 radius.

SECTION 2 - ACCEPTANCE CRITERIA

2.1 PERFORMANCE

The contractor shall subject the motor(s) to the following test and analysis cycle:

(1) Room temperature calibration

Develop motor speed, torque and efficiency characteristics per industry accepted standards. Tests to be performed at room temperature.

(2) High temperature no/load

Run motor for 8.0 hours in an oven with an average ($\pm 10^{\circ}\text{C}$) temperature of 275°C at no/load, and rated voltage. Current to be monitored and recorded during test.

(3) Post test calibration

Upon completion of the hot no/load test, repeat room temperature calibration.

(4) High temperature performance

Using industry accepted standards and results from tests (1), (2) and (3) listed above, estimate the performance of the motor at rated load and temperature. "Acceptable performance" is that defined in paragraph 1.1 of this specification.

2.2 ENVELOPE

Conformance to the envelope requirements, paragraph 1.3, shall be met by industry accepted inspection procedures done at room temperature. These inspection records shall be provided with each motor.

SECTION 3 - QUANTITY/DELIVERY

3.1 QUANTITY

This procurement is for two (2) motors.

3.2 DELIVERY

Delivery is to be made 20 weeks after receipt of order.

SECTION 4 - LIAISON

4.1 LIAISON

The contractor shall provide approximately one (1) hour of time every six weeks to meet with the cognizant contractor engineer to review the project status. These meetings will be at the contractor's facility at a mutually agreeable time.

ALL-8

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES						SDC SYSTEM DEVELOPMENT CORPORATION 3300 COLORADO AVENUE SANTA MONICA, CALIFORNIA 90406	
TOLERANCES ON						TITLE	
BASIC DIM	2 PLACE DECIMALS	3 PLACE DECIMALS	DRAFTSMAN	P-H	4-2	MOTOR INTERFACE	
UP TO 6	± .02	± .005	CHECKER			CODE ID NO.	REV
ABOVE 6 TO 24	± .03	± .010	ENGR			14155	A
ANGULAR DIM TOLERANCE ± 1°			CNDR			DWG NO.	TNT-41-018
			APPROV.			SCALE: —	WT. —
			APPROV.			SHEET 1 OF 2	

ALL-9

#4-40 UNC-2B x .31 MIN DEEP THREAD
EQUALLY SPACED ON 3.25 DIA B.C.
± .005 DIA.
TYP BOTH ENDS

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES			MATERIAL 17-4 ST. STEEL (50 KSI)		SDC SYSTEM DEVELOPMENT CORPORATION 2500 COLORADO AVENUE SANTA MONICA, CALIFORNIA 90405	
TOLERANCES ON			DRY TERN	PH	4-4	TITLE MOTOR INTERFACE
BASIC DIM	2 PLACE DECIMALS	3 PLACE DECIMALS	CHECKER			CODE ID NO. 14155
UP TO 6	± .02	± .005	ENGINEER			SIZE A
ABOVE 6 TO 24	± .03	± .010	APPROV.			DWG NO. TNT-41-018
ANGULAR DIM TOLERANCE ± 1°			APPROV.			REV -
SCALE: —			WT. —		SHEET 2 OF 2	