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DESIGN AND DEVELOPMENT OF A WATER-FLOODED SCREW COMPRESSOR PACKAGED AIR SUPPLY SYSTEM

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SYNOPSIS The first phase of the development program successfully demonstrated the technical feasibility of water-flooded operation of a helical screw compressor for delivery of non-lubricated air at gauge pressures up to 896 kPa in a single stage. A compressor configuration evolved that performed successfully for prolonged operating periods at a performance level (W/m^3) competitive with other types of equivalently rated machines. Satisfactory solutions to anticipated problems of (1) lubricating oil-air isolation, (2) construction materials selection, and (3) solids deposition within the compression zone were demonstrated.

During the second phase a commercial prototype, packaged, air supply system was designed and built to operate at the optimum performance point determined during the dynamometer testing of the Phase One unit. This unit then successfully completed 2700 hours of operation under typical field conditions of load-no load and start-stop. A satisfactory solution was demonstrated for air-water separation on this unit.

Following the successful operation of the commercial package in Phase Two, the U.S. Navy requested that Ingersoll-Rand design a unit to meet Naval Specifications for a shipboard air supply package, delivering 47.2 l/s at a discharge gauge pressure of 862 kPa. The Navy prototype unit was designed and developed under Phase Three of the program by the Engine Process Compressor Division, with the assistance of the Ingersoll-Rand Research Center and Svenska Rotor Maskiner AB. Two prototype units were built and tested for a total of 8000 hours. These units basically met all the Navy requirements except for a high power requirement resulting from over capacity.

Phase Four of the program was the final design and development of the Navy Production Package. The production unit satisfied or exceeded all Naval specifications with respect to performance, noise, shock and reliability. Further, the unit exhibits substantial reductions in size and weight over the equivalent reciprocating compressors presently in service.

1.0 INTRODUCTION

1.1 The stimulus for this program was the desirability of adapting the screw compressor to new market applications. One application envisioned was the requirement of many industries for a non-lubricated (oil free) air supply. If the screw compressor could be operated with water, rather than oil flooding, such markets could be reached. Major advantages for such a machine concept included:

- Growing industry preference for rotary machinery
- Combined air/water cooler
- Smaller compressor
- Smaller number of components

1.2 The development program was initiated in 1970 to demonstrate the technical feasibility of water flooded operation of the screw compressor. An air-end configuration evolved which was subjected to performance evaluation and durability testing. This testing demonstrated technical feasibility and that competitive performance was attainable. A commercial, packaged air supply system was designed utilizing the optimum operating conditions. U.S. Navy interest in this activity resulted in a program to develop a shipboard-rated unit. This program has now progressed to the point of field evaluation of several units on U.S. Navy vessels.

1.3 This paper presents the various phases of activity in the development of the water-flooded screw compressor from initial concept through qualification for shipboard service.

2.0 PHASE 1

2.1 Compressor design

2.1.1 A vertically split experimental compressor was designed embodying the mechanical design and operating features shown in Tables I and II Phase I. Within the basic design, provisions were made to permit compressor performance evaluation with three different water injection techniques; cusp injection (standard drilled orifices), helix injection (drilled orifices or solid cone sprays), and inlet injection (single solid cone spray). An exploded view of the compressor is shown in Figure 1.

2.1.2 One of the primary design points was whether to use timing gears to prevent inter-rotor contact or drive directly through the rotors like oil-flooded units. The analysis indicated that water would not provide adequate lubrication between rotors and that timing gears were necessary.

2.2 System selection process

2.2.1 At the outset it was necessary to consider whether an open or closed-loop water system represented the best approach. In the open system, raw water would be injected into the compression cavity, discharged with the air, separated and discarded. In the condensate recycle system, water would be injected during the compression process, discharged with the air, separated, cooled and then re-injected, thus continually recirculating through the system.

2.2.2 A number of problems were anticipated which were common to both approaches. Major concern centered around: (1) solid deposits from the water (hardness) binding the machine, (2) isolation of injected water from bearing lubricating oil, (3) achievement of competitive performance, and (4) protection from corrosion during operation and down time.

2.2.3 A number of important advantages favored the ultimate selection of a raw water, condensate recycle system. First, the volume of water containing dissolved solids to which the internal compressor cavities would be minimal reducing the tendency for solids to precipitate out. Secondly, with minimal or no direct water consumption there would probably be no need for water conditioning equipment to remove hardness. Only the problem of corrosion of internal parts then would need to be addressed.

2.3 Experimental test system

2.3.1 The test system used for performance evaluation of the compressor is shown in Figure 2. The compressor was driven by a 44.7 KW electric dynamometer through a gearbox, having a step-up gear ratio of 2.87:1. The dynamometer and gearbox were direct connected through a shear pin gear coupling. The gearbox output shaft was connected by a gear coupling to the male rotor shaft for the compressor drive. This arrangement permitted male rotor shaft speeds to 167 Rev/s (the desired upper speed limit) at the dynamometer continuous-operation limiting speed of 60 Rev/s.

2.3.2 Water for injection was stored in a vented tank located such as to provide adequate positive inlet pressure to a 2-stage centrifugal pump. The pump along with a throttling valve provided the flexibility to control water injection flow-rate independently of compressor operating speed and discharge pressure. Water was delivered to the compressor via a float-type flowmeter and filter.

2.3.3 Filtered air entered the compressor through a conventional air flow measuring orifice. After compression and mixing with the injected water, the discharge flow passed through a swing check valve (to limit post-shutdown reverse motoring) and entered the tube side of the water-cooled aftercooler. Separation of water and air took place in the separator with the water then being returned to the tank through traps for reuse. Air was released to the atmosphere through a pair of throttling valves piped in parallel.

2.4 Performance and durability testing

2.4.1 Performance evaluation tests were made to establish the capability of the compressor with the three methods of water injection. The helix injection method (with spray nozzles) was found to be slightly better and only the results of these tests are presented in Figure 3. On the basis of these results, durability testing was initiated with a motor drive at the optimum performance point of approximately 83.3 Rev/s and .95 l/s of water. Over four thousand hours of operation were achieved without incident.

2.5 Results

2.5.1 Water flooded operation of a screw compressor over prolonged periods of operation in a raw water condensate recycle system was demonstrated to be feasible.

2.5.2 Solid deposits from the raw water did not present operational problems with the condensate recycle system. (Water hardness range 60 to 260 RPM).

2.5.3 While oil-air and oil-water isolation was demonstrated with air injected labyrinth seals, they are sensitive to air injection pressure.

2.5.4 Performance of a water flooded screw compressor was shown to be competitive with other equivalently rated machines.

2.5.5 The water flow-rate requirement was found to be approximately twice that of an equivalent oil flooded machine for optimum performance.

2.5.6 Corrosion resistant materials are required for the rotor lobes and all materials surrounding the rotor lobes.

2.5.7 With respect to performance, helical water injection was found to yield slightly better performance than either the cusp or inlet type of water injection.

2.5.8 There is negligible difference in performance between the use of drilled orifices or nozzles with helical water injection.

2.5.9 Optimum performance was found to be at approximately 83.3 Rev/s (equivalent to a rotor tip velocity of 27 m/s).

3.0 PHASE II

3.1 Compressor modification

3.1.1 A new compressor was designed embodying a splash lubrication system for timing gears and bearings in place of the previous pressure system. Also, carbon ring face seals were substituted for the labyrinth seals at the discharge end of the machine to provide additional margin against water migration into the oil sump. To achieve optimum speed with a standard 60 Rev/s electric driver, female rotor drive was utilized.

3.2 Package design

3.2.1 In view of the success with Phase I, the decision was made to design a commercial integrated air-supply package for endurance testing under simulated field conditions.

3.2.2 For the package we drew heavily on the IR PAC-AIR line of oil-flooded, screw compressor industrial air supply concept, eliminating all oil-related components while maintaining many of the controls and the unloading system. With the integrated package, the system air pressure was utilized to pressurize the water for injection. A special ASME Code-rated tank was designed as a combination receiver-separator, demister and aftercooler. Figure 4 shows the completed system with the exterior soundproofing shell removed.

3.2.3 The nominal system design specifications are given in Table I and II Phase II.

3.3 Testing

3.3.1 Initially, testing centered around system debugging and the resolution of certain mechanical problems. Once it was established that compressor operation was normal and that all features of the control systems were operative, as demonstrated by limited endurance testing in both the load/no-load and automatic start-stop modes, the system performance evaluation tests were made. System performance was evaluated over the discharge gauge pressure range from 552 to 827 kPa.

3.3.2 After system start-up, control system debugging and performance testing, it was the objective of the program to accumulate operating time as quickly as possible. To this end the system was automated to permit unattended, continuous automatic cycling between the loaded and unloaded modes of operation, providing a close simulation of actual service use.

3.3.3 Approximately 2700 hours of operation were accumulated on the development package, with 98% of this time occurring in the loaded mode. The system entered the unloaded mode over 1500 times and there were 164 start/stop cycles.

3.3.4 It is interesting to note that in the colder months of the year, makeup water was required to replace water vapor escaping from the system when the relative humidity of the intake air varied from approximately 17 to 30%. During the summer months when the relative humidity ranged from 40 to 55%, the system generated water which periodically was automatically dumped from the system. The result of the latter situation was to progressively dilute the initial raw water with distilled water extracted from the air. Accordingly, the hardness of the water in the package was gradually reduced.

4.0 PHASE III

4.1 U.S. Navy oil free compressor program

4.1.1 During the late 1950's, the U.S. Navy had begun a program for the development of oil-free compressors for shipboard service. While the program was instituted to develop high pressure compressors, it was later expanded to inlcude low pressure machines. The outcome of the program was that both low and high pressure reciprocating oil-free compressors were installed aboard ships. The Navy then directed its attention to other types of compressors that could provide advantages over the reciprocating type for shipboard service. One concept investigated was the use of the helical screw compressor.

4.2 Performance objectives

4.2.1 After successful Phase II tests of the compressor package, the Navy personnel were invited to witness operation of the machine. After several meetings to determine the Navy performance requirements, a proposal was submitted to the Navy for two units. The units were to be designed to meet the operational design features shown in Table II Phase III.

4.3 Compressor design

4.3.1 Design of the unit commenced in the spring of 1973. Since Ingersoll-Rand is a licensee of Svenska Rotor Maskiner AB for the helical screw compressor, it was decided to utilize their design experience. Working closely with SRM, the performance objectives were analyzed to arrive at what was considered the optimum air end configuration.

4.3.2 To meet the stringent noise levels, a low rotor tip velocity was selected with direct drive through the male rotor shaft. This would reduce the fluid-borne noise and, by driving through the male rotor, the timing gears would only transmit approximately 12 percent of the total horsepower required, thus minimizing timing gear noise. A rotor diameter of 112 mm was selected with a rotor tip velocity of 20.7 m/s and a length to diameter ratio of 1.5 (rotor length 168 mm). The electric motor direct drive set the male rotor speed at 58.8 Rev/s and a female rotor speed of 39.2 Rev/s. Complete compressor mechanical design features are summarized in Table I Phase III.

4.3.3 Bearing loads were calculated and various arrangements were analyzed for maximum life. The arrangement selected located the thrust bearings at the inlet end, with the timing gears and the male rotor drive at the discharge end. For maximum reliability, mechanical face seals were selected to seal and isolate the air-water system from the bearing lubrication system. Due to the requirement for inclined operation, various bearing lubrication schemes were analyzed and a disc slinger system was adopted. This provided a simple, trouble-free system, completely self-contained in each end housing.

4.3.4 Since the water injection points had not been fully established on the research prototype, the Navy unit incorporated three points for injection. Cusp injection at the rotor mid-point in the bottom cusp, female rotor injection which had proven successful in oil-flooded units, and inlet injection. In addition, water was injected into the air-water seal cavity to provide cooling for the seal and to seal the rotor end faces.

4.3.5 To meet corrosion resistance requirements, stainless steels were selected for the rotors and housings. Horizontal and vertical sectional views of the compressor are shown in Figure 5.

4.4 Water injection system

4.4.1 Next, the water injection system was analyzed. Water injection rates were estimated at .42 to .50 l/s. The system would require a storage tank, air-water separation device, cooler, controls, valves and interconnecting piping.

4.4.2 Two locations were considered for the cooler. First, as an aftercooler to cool the air-water mixture; and second, as a water cooler only. Analysis showed that using the cooler for the water only provided better heat transfer and required a smaller cooler size.

4.4.3 The major system problem centered around an efficient device to separate the water from the air and an adequate reservoir that could be controlled to an operating level under ship roll and pitch conditions. Final design of the separation unit provided a swirl configuration with velocity reduction, change of direction, and then increased velocity through a coalescing element. The separator was designed into the top section of a cylindrical tank mounted vertically. The bottom of the tank provided the reservoir and water level controls mounted in the center of the tank to minimize the effect of ship roll and pitch. The air and water flow diagrams are shown in Figures 6 and 7, respectively.

4.5 Controls

4.5.1 Original Navy specifications for compressor control required two modes of operation. One, start-stop; and second, constant speed control. While constant speed operation is practical on reciprocating compressors with inlet valve unloading, it is very inefficient on a fixed port helical screw compressor. It was decided to use only the start/stop mode, either manual or automatic for this unit. In the automatic mode, the unit would unload by use of an inlet throttle for a period of 10 minutes. If air receiver pressure did not drop below a set point within the 10 minutes, the unit would shut down. The unit would start again when the low set point was reached.

4.5.2 Control of water supply for the water injection system consisted of a four-switch magnetic float control in the reservoir. The two center switches provided automatic water dump or add, whichever was required. The top switch provides for a high water level shutdown and the bottom switch for low water level shutdown.

4.5.3 In the event of loss of injection water to the compressor, a high discharge temperature shutdown control is located at the compressor discharge flange.

4.6 Package design

4.6.1 The next step was the design of the package when all the components were combined into a compact, light-weight unit meeting the operational design features shown in Table II Phase III. While the air end had been designed to meet the shock requirements of MIL Spec 901G, there remained a question as to the effect of the shock on the rotors during operation. Due to the close clearances required to obtain maximum efficiency, it was considered almost certain that under maximum shock levels the rotor would deflect and result in inter-rotor and rotor-to-housing contact. Taking this into consideration, various package arrangements were considered. Mounting the helical screw compressor and the drive motor on one subbase provided many advantages. By mounting the subbase on the main base with rubber isolation mounts, both shock and noise levels were reduced. With the units mounted higher, it provided flow of injection water back to the reservoir and improved accessibility for day-to-day maintenance.

4.6.2 The main base and subbase utilized box shape structural members to obtain maximum strength to weight ratios. When completed, the package attained a substantial reduction in size and weight over the reciprocating units. See Table III, Column A & Phase III.

4.7 Preliminary compressor test

4.7.1 The first helical screw compressor was completed in the spring of 1974 and underwent preliminary testing at the Svenska Rotor Maskiner Works in Stockholm. The test indicated that the estimated water injection rates of .42 to .50 l/s were low and that the unit attained maximum performance with rates of .63 to .75 l/s.

4.7.2 Volumetric efficiency was higher than anticipated and resulted in capacities of 50 to 54 l/s along with concurrent high horsepower. Minor modifications were made to the inlet porting which provided some reduction in capacity and horsepower.

4.7.3 Overall, the testing indicated that:

1. Use of any one of the water injection points or combination thereof had little effect on performance.

2. Water injection into the air-water seal cavities had a significant effect on performance.

3. At constant injection rates, discharge temperature was solely dependent on water injection temperature.

4.7.4 Performance characteristics. Capacity, specific power and discharge temperature at various water injection rates are shown in Figure 9. After approximately 100 hours of testing, the unit was dismantled and inspected for any signs of distress to the mechanical components. The only problem indicated was high wear rates of the air-water mechanical face seals. A material change was made from a chrome to a chrome oxide faced seal washer. Subsequent testing showed minimum wear of the new material.

4.8 Package test

4.8.1 The unit was then shipped to the Ingersoll-Rand plant in Painted Post, New York, where it was assembled in the prototype package. The complete package testing started in the summer of 1974 and continued into 1975. During the package test, the base performance of the helical screw unit was approximately the same as indicated during the SRM tests. The most significant development from the test was the point of water injection on air-borne and structure-borne noise levels. While the point for water injection had little or no effect on base performance, it did have a significant effect on noise. These tests resulted in the inlet being the overall optimum point for injection with the spray pattern oriented toward the female rotor. The prototype unit met all the design objectives except for maximum power consumption.

4.9 Navy evaluation

4.9.1 The Navy decided to proceed with their evaluation program on the basis that production units would be modified to correct the over-horsepower condition. To date, the Navy has operated one unit approximately 5000 hours and the second 3000 hours. Their preliminary evaluation of the unit was that the unit provided excellent performance and offered significant size, weight and noise advantages over corresponding reciprocating compressor.

4.9.2 The Navy, while in the process of approving the unit for shipboard service, decided that the production package should also incorporate a dehydration unit and receiver.

5.0 PHASE IV

5.1 Modifications

5.1.1 The package was modified to incorporate a refrigerated dehydrator and receiver along with a minor change in the air end rotor L/D ratio to correct the overpower condition. The final package physical characteristics are shown in Table III, Phase IV with the overall arrangement shown in Figure 8.

5.2 Testing

5.2.1 The unit then underwent final performance, noise, shock and vibration testing. The air end rotor L/D modification produced the required power reduction as shown in Figure 9.

5.2.2 The structure-borne and air-borne noise levels are shown in Figures 10 and 11 along with the Navy specified requirements and typical reciprocating compressor levels.

5.2.3 Shock testing was performed for both hard and soft mounting of the main base. While the design of the subbase to main base mounting with isolation mounts did not reduce the shock levels on the compressor in the soft mount test, it reduced the shock level by 50% in the hard mount test.

5.3 Installation

5.3.1 The first production units are being installed aboard a new class of Navy ships.

6.0 CONCLUSIONS

6.1 Sustained operation of a water-flooded screw compressor package delivering NL air at a performance level competitive with other types of equivalently rated machines was demonstrated.

6.2 Mechanical face seals provide a more reliable seal for lubricating oil-water isolation.

6.3 A self-contained splash lubrication system provides adequate bearing and timing gear lubrication.

6.4 All system and compressor components that come in contact with the compressor injection water should have corrosion protection.

6.5 Off periods of as long as 2 months were not detrimental to equipment or subsequent compressor and systems operation.

6.6 The Navy production package realized significant reductions, over comparable reciprocating compressor package, in:
(1) Air-borne and structure-borne noise levels, and (2) Physical size and weight.

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T.D. O'Leary, Designer, I-R EPCD

Table I - Compressor Mechanical Design Features

Research Program

	Phase I	Phase II	Phase III	Phase IV
Rotors:				
Profile	Asymmetrical with 300° wrap angle			
Lobe Ratio	6:4 Female to Male			
Diameter - mm	102	102	112	112
Length/Dia. Ratio	1.25	1.25	1.5	1.406
Bearings:				
Thrust	Angular contact double row ball			
Support	Deep groove radial ball		Roller	
Lubrication	Pressure	Splash	Splash	Splash
Seals:				
	Labyrinth Air injection	Mechanical Face	Mechanical Face	Mechanical Face
Timing Gears:				
	Straight Spur	Straight Spur	Helical	Helical
Porting:				
	Variable	Variable	Fixed	Fixed
Materials:				
Rotors	AISI 416SS			
Housings	AISI 416SS	AISI 416SS	AISI 410SS	AISI 410SS

Table II - Operational Design Features

	Research Program		U.S. Navy Program	
	Phase I	Phase II	Phase III	Phase IV
Built-in Pressure Ratio	6.0-13.0	8.0	8.5	8.5
Speed Range Male Rotor - Rev/s	0-167	88.5	58.8	58.8
Discharge Gauge Pressure - kPa	896	690	862	862
Inlet Capacity & Discharge Gauge Pressure-1/s	Vari.	44.7	47.2	47.2
Driver: KW	44.7	18.6	22.4	22.4
Rev/s	0-60	59	58.8	58.8
Injection Water:				
System Capacity - 1	---	95	19	19
Flow Rate 1/s	0-1.58	1.07	.42-.50	.63
Pressure - kPa	Vari.	520	550 Minimum	550 Minimum
Reliability	----	----	--------20,000 Hours------	
Ship Roll & Tilt	----	----	----Up to 45 Degrees ----	
Shock	----	----	-----Mil. Std. 901C------	
Noise	----	----	-----Mil. Std. 740 ------	

Table III - U.S. Navy Package Physical Characteristics

		Phase III	Phase IV
	Equivalent Navy Reciprocating	Prototype Navy Screw	Production Navy Screw
Length mm	1575	1016	1390
Width mm	1372	1194	1359
Height mm	1753	864	1200
Deck Area m^2	2.16	1.21	1.89
Volume m^3	3.77	1.05	2.27
Weight Kgm	2177	816	998

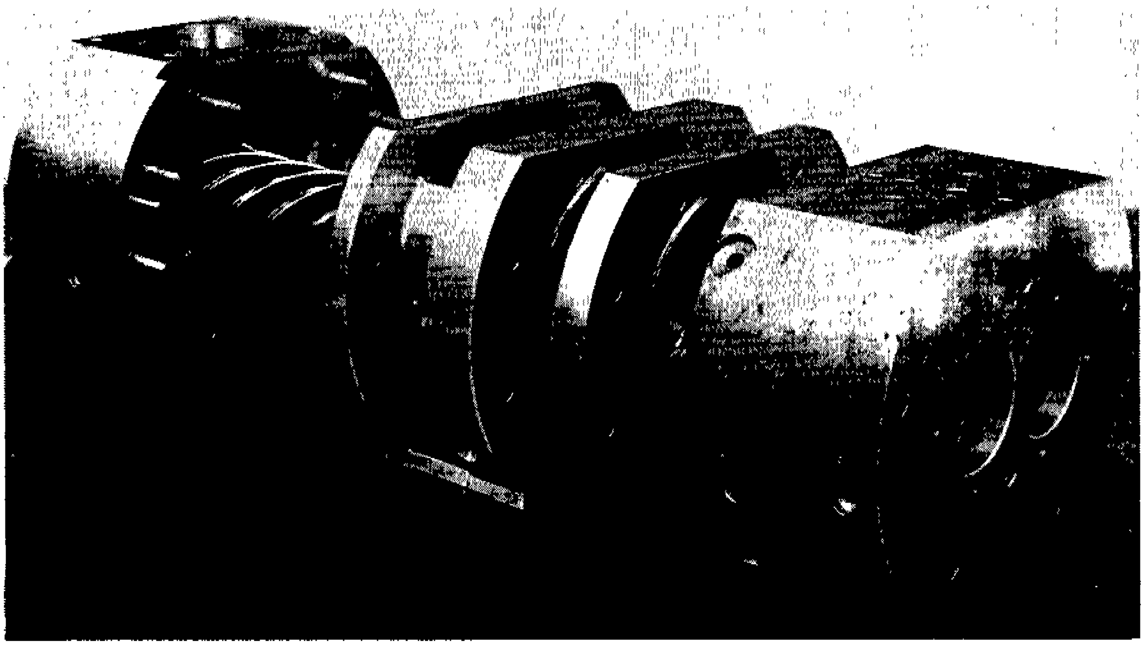

FIGURE 1: EXPLODED VIEW OF DEVELOPMENT COMPRESSOR

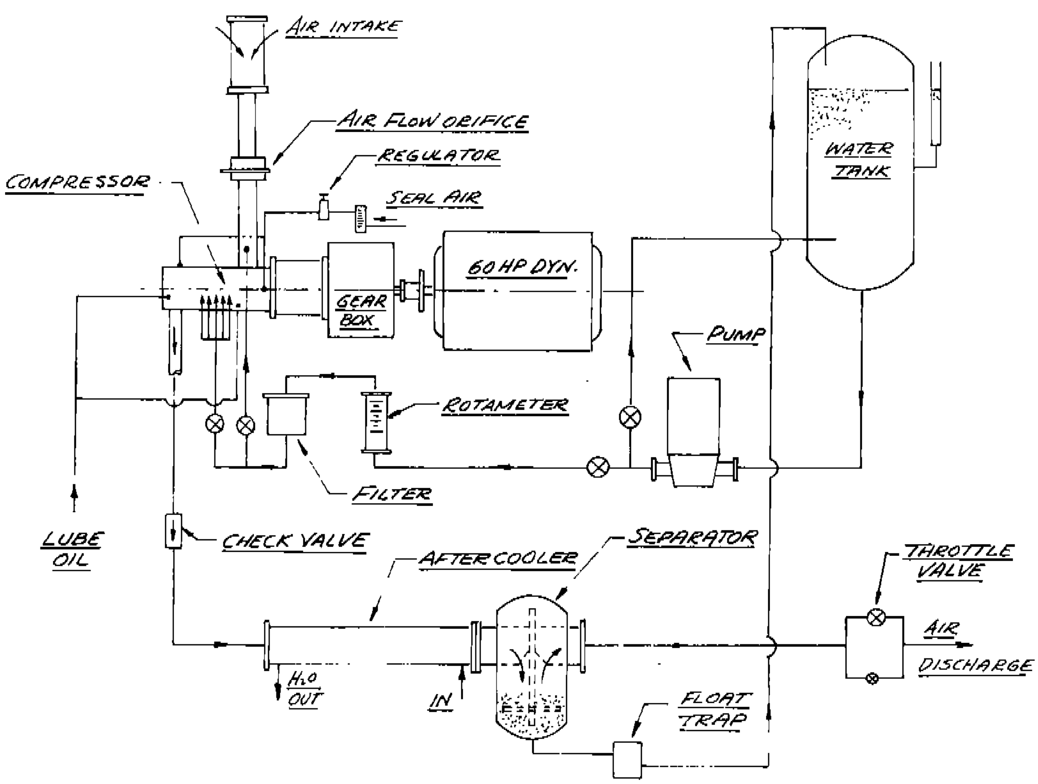

FIGURE 2: EXPERIMENTAL SYSTEM

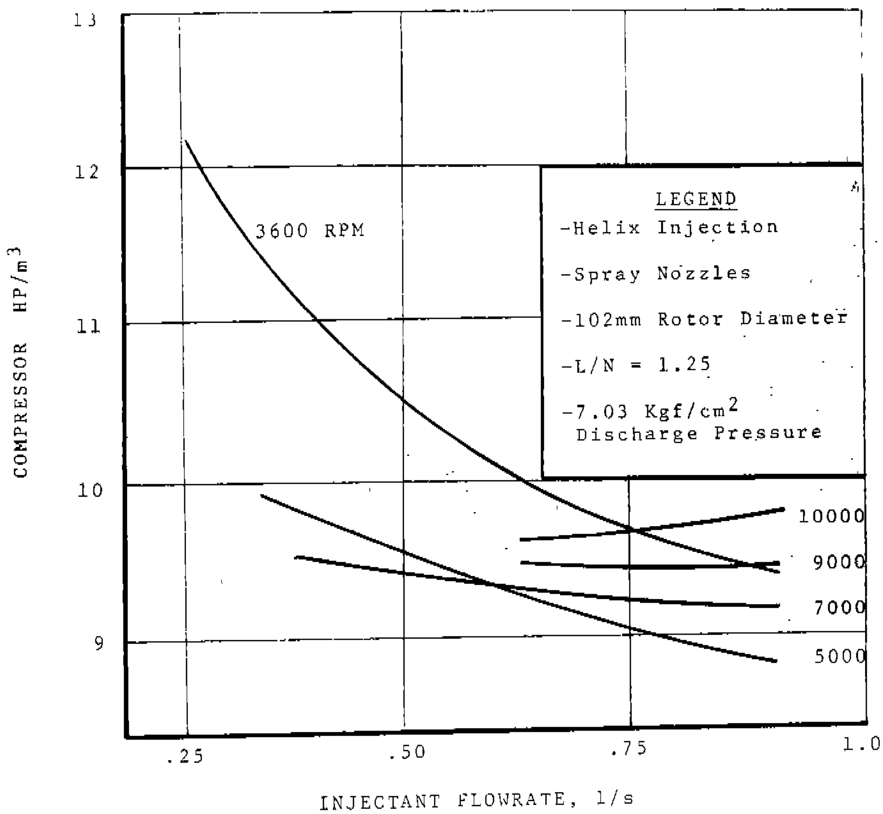

FIGURE 3: COMPRESSOR Hp/m^3 VS INJECTANT FLOWRATE

FIGURE 4: GENERAL VIEW OF DEVELOPMENT PACKAGED AIR SUPPLY

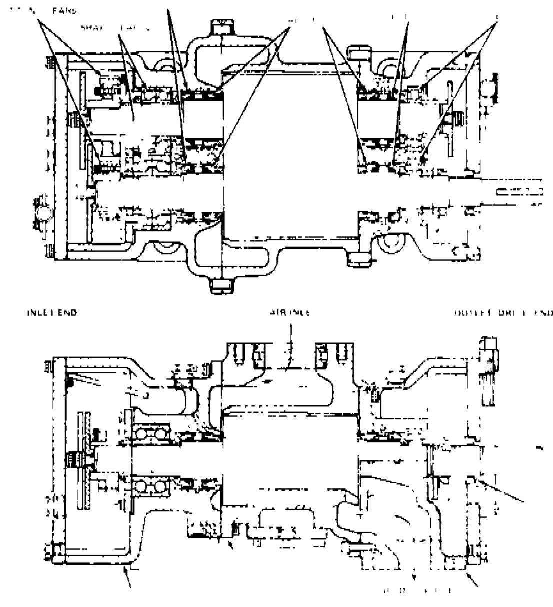

FIGURE 5: SCREW COMPRESSOR (HORIZONTAL & VERTICAL SECTIONS)

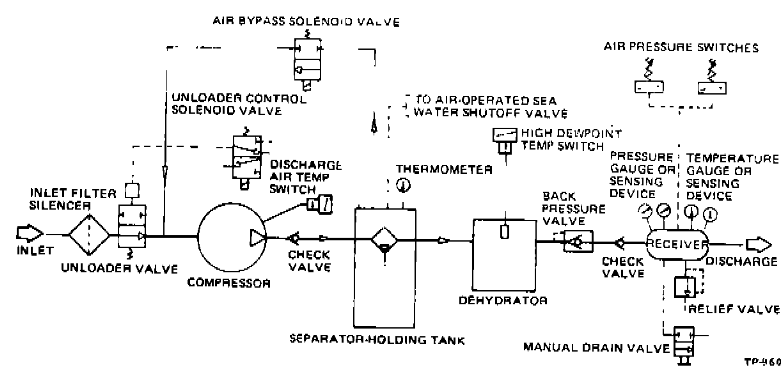

FIGURE 6: AIR FLOW DIAGRAM

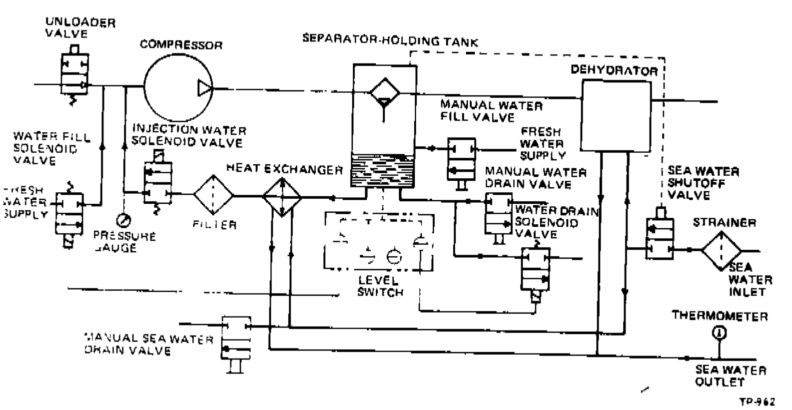

FIGURE 7: INJECTION WATER & SEA WATER DIAGRAM

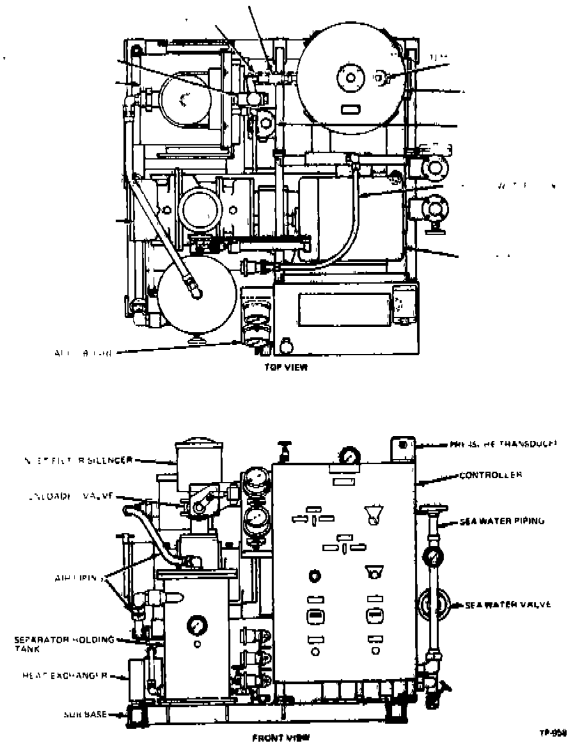

FIGURE 8: PHASE IV PACKAGE

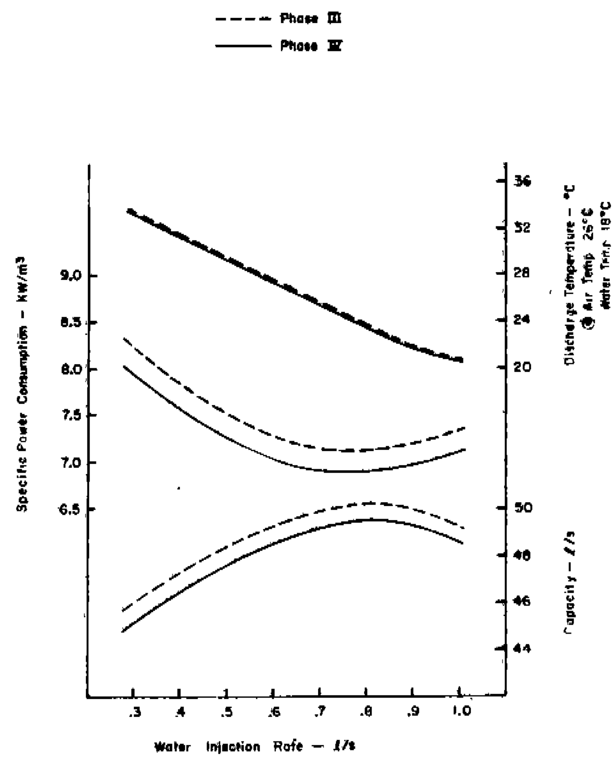

FIGURE 9: PERFORMANCE CHARACTERISTICS

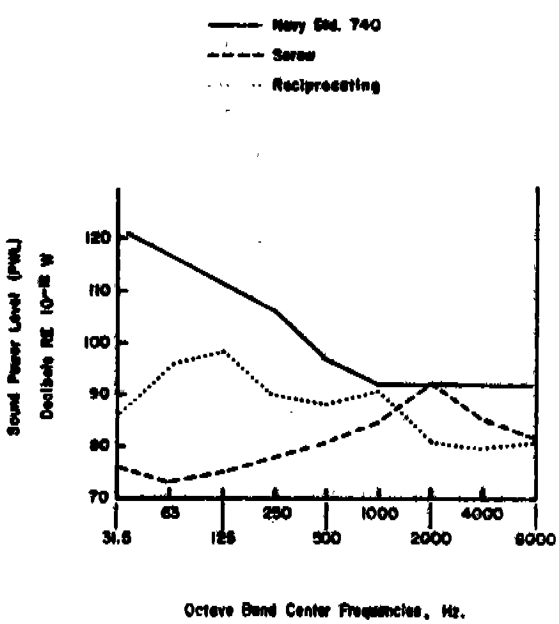

FIGURE 10: AIR-BORNE NOISE LEVELS

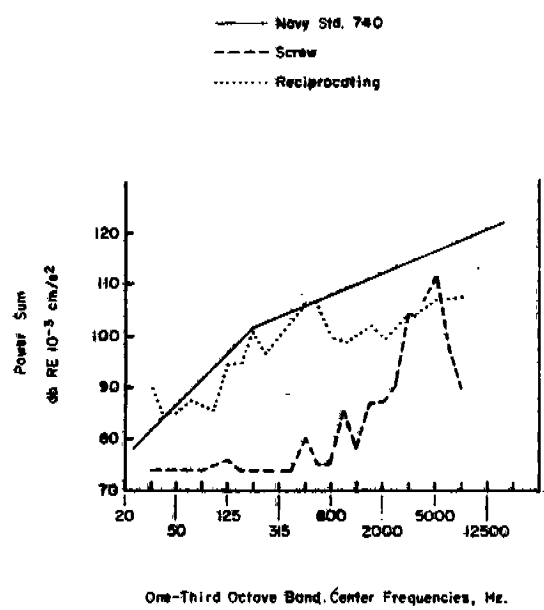

FIGURE 11: STRUCTURE-BORNE NOISE LEVELS