OAK RIDGE NATIONAL LABORATORY

operated by

UNION CARBIDE CORPORATION

for the

U.S. ATOMIC ENERGY COMMISSION

UNION CARBIDE

ORNL - TM- 63

COPY NO. - 69

DATE - Oct. 9, 1961

DEVELOPMENT OF CENTRIFUGAL COMPRESSORS

I. K. Namba

Abstract

Coolant flow for gas-cooled in-pile loops must be supplied during irradiation test runs. A centrifugal compressor has been designed and developed for circulating helium at volume flows from 75 to 250 acfm at compressor suction conditions of 400 psi and 600°F. The compressor using grease-lubricated ball bearings has operated satisfactorily for a total of 3500 hr.

DEVELOPMENT OF CENTRIFUGAL COMPRESSOR FOR HELIUM CIRCULATION

I. K. Namba

Introduction

A need existed for a compressor to circulate helium gas at relatively elevated temperatures in an in-pile irradiation facility and the circulation was required to be supplied reliably and continuously for extended periods of time. Maintenance-free operation was a requirement, since the compressor would be hermetically sealed in a pressure vessel (together with all of the other loop components) and immersed in the reactor pool water. Initial studies showed that it would not be practicable to purchase a compressor from a commercial source, because the very stringent space limitations imposed on the test loop by space availability in the reactor pool made it necessary to design all of the components inside the pressure vessel as a single package. The compressor design was therefore undertaken at Oak Ridge National Laboratory.

A centrifugal impeller and a vaned diffuser were chosen for the aerodynamic portion of the compressor. The impeller and the rotor of the drive motor were mounted on a single shaft which was supported on grease-lubricated ball bearings.

Aerodynamic performance data for the compressor were obtained with helium at design suction conditions of 400 psi and 600°F. An endurance test of 3500 hr duration at design suction conditions was conducted to evaluate the integrity of the bearing housing design and the longevity of the commercially available grease-lubricated bearings.

Description of Compressor

The principal components of the centrifugal compressor (Fig. 1) are the rotary assembly and the stationary assembly. The rotary element, consisting of shaft, impeller, and motor rotor, is supported on grease-lubricated ball bearings. The inner diameter of the shaft, and the stationary assembly (which consists of the front and rear bearing mounts and the motor stator), are water cooled for protection from the effects of suction temperatures approaching 600°F. The need for forced cooling and heat sinks is based on two reasons: 1) the grease lubricant can be damaged by temperatures in excess of 200°F; and 2) the electrical insulation may be damaged at hot-spot temperatures exceeding 350°F.

The water cooling system for the main housing (Fig. 1) consists of a double spiral channel with metal-to-metal fitted cover shells and plates. The shaft is bored to accommodate a "cold-finger" assembly to provide an additional heat sink for the ball bearings. With appropriate connections, the main housing coolant passages are series connected with the cold-finger assembly.

UNCLASSIFIED
ORNL-LR-DWG 55754

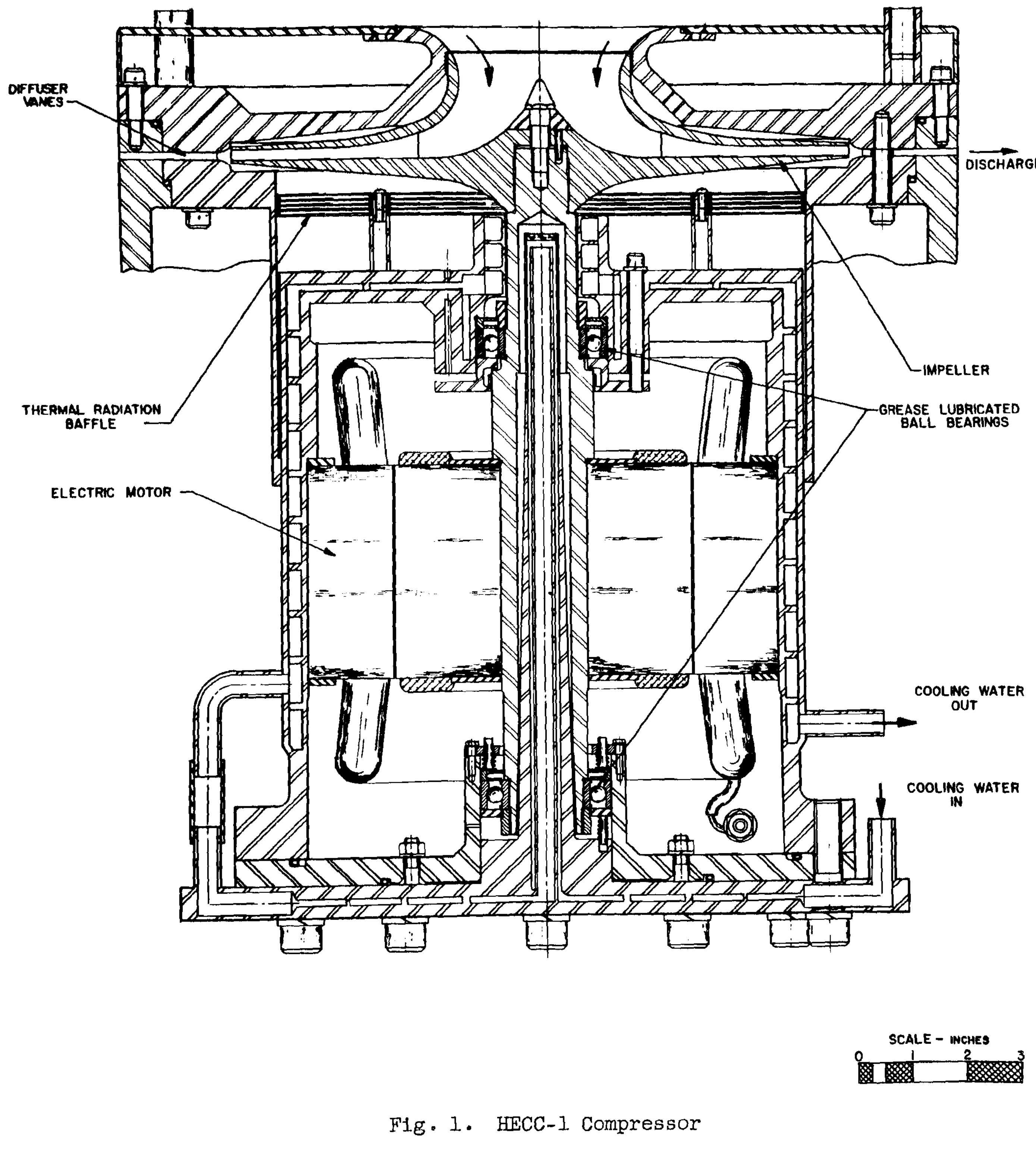

Fig. 1. HECC-1 Compressor

In order to use commercially available ball bearings[1] and grease lubricant, the temperature in the bearing housings was maintained in the range of 125 to 150°F. To achieve reliable and maintenance-free life of the bearings for extended periods of operation, the concentricity of the bearing housings was held to a tolerance of 0.0005 in. TIR. For the vertical orientation of the compressor, a 50-lb preload[2] was imposed on the bearings.

The rotary element, consisting of shaft, motor-rotor, and centrifugal wheel, was dynamically balanced to 0.2 g-in. For the aerodynamic configuration, a fully shrouded impeller and a vaned diffuser were designed using conventional aerodynamic principles[3] to be capable of operation at all conditions from ambient up to and including design operating conditions.

A thermal radiation barrier and water-cooled heat sink were interposed between the hot gas surfaces and the upper bearing housing. A thermal sleeve was provided on the housing to minimize the thermal gradient between the diffuser (600°F) and the water-cooled main housing.

The drive unit for the compressor is a four-pole shell-type electric motor suited to the configuration of the compressor. The motor, rated at 20 hp at 400 cps, has a constant torque between frequencies of 200 to 400 cycles when the applied voltage is proportional to frequency (0.52 volts per cycle).

The outside diameter of the motor stator was limited by the overall permissible diameter of the test loop. It has since been determined that it would be possible to increase the rating of the motor considerably by using a larger stator, while retaining the same rotor dimensions.

It was originally planned to install two compressors (one above the other) in the in-pile test loop. Gas flow from the vaned diffusers was to be turned 90° downward, and guided either to the lower compressor or to the other parts of the system, through a cylindrical annulus parallel to the axis of the shaft. In addition to the diffuser vanes, it was planned to incorporate straightener vanes in the cylindrical annulus to permit additional conversion of velocity head to static head. The inside diameter of the cylindrical annulus formed a shielding cannister to minimize the flow of heat from the gas in the annulus to the compressor coolant system.

With the exception of the electric motor, the major components of the compressor were fabricated from Inconel, which has approximately the same coefficient of expansion at operating temperature as the steel used in the motor rotor and stator, and in the bearings.

Test Facility

Tests were accomplished in a closed-circuit test loop (Fig. 2), which consisted of the compressor housed in a pressure vessel and the necessary

UNCLASSIFIED
ORNL-LR-DWG 59199

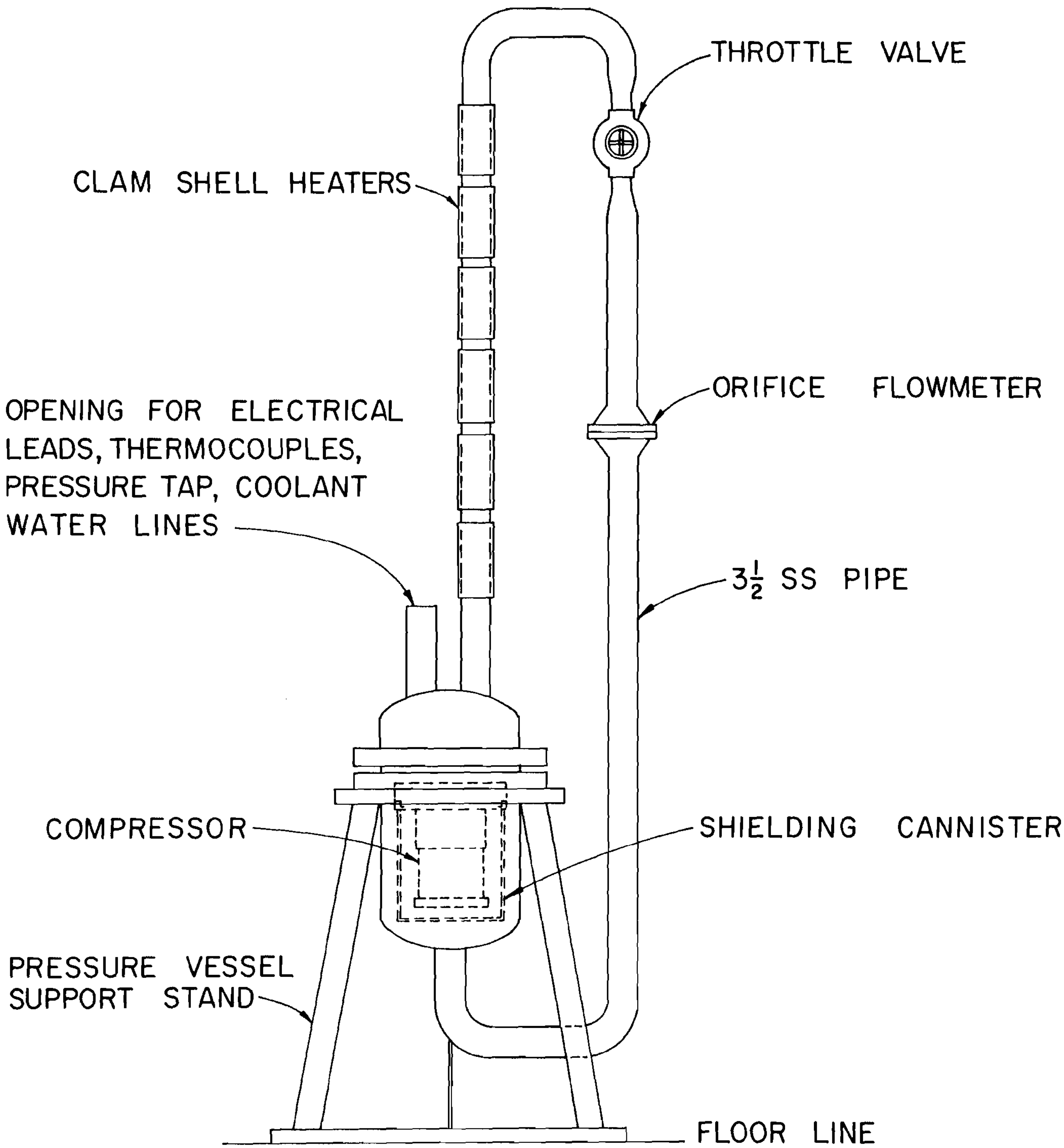

FIG. 2 HELIUM COMPRESSOR HOT TEST LOOP

piping and instrumentation to obtain performance data. The compressor was mounted with the shaft vertical, and with the impeller end at the top. The pressure vessel, being an integral part of the loop system, was seal-welded at the flange closure to minimize system gas leakage.

An external heat source to provide design operating temperatures was supplied from banks of clam-shell heaters, regulated by variac control. The compressor coolant supply was provided by appropriately manifolding the water connections through the pressure vessel. The power leads and instrumentation connections (thermocouples and pressure taps) were similarly made.

Development Procedure

In order to ascertain the integrity of a developmental compressor, it was decided to use a cold shakedown test followed by a hot shakedown run. The cold shakedown test showed that the bearings, grease seals, thermocouples, and coolant system were functioning properly. The power characteristics of the drive motor and bearing-housing temperatures were noted for evidences of abnormal behavior.

The hot shakedown test is the logical continuation of the cold shakedown run to check the mechanical reliability and performance of the compressor at an elevated gas temperature environment by observing power traces, bearing-housing thermocouple readings, vibration and noise characteristics, and to obtain aerodynamic data.

Development Problems and Testing

One of the problems common to rotating equipment is running clearances. During the cold shakedown period, two incidents of inadequate running clearances occurred: 1) the rotor-stator air gap was too small; and 2) the shaft-impeller assembly rubbed against the labyrinth seal.

When the first attempt was made to start the compressor, it was found that it could not be accelerated to a speed of much more than a few thousand rpm. Consultation with ORNL electrical engineers and with the manufacturer of the motor indicated that the difficulty was because of a combination of two factors: 1) the generation of harmonics in the motor, which caused a substantial cusp in the speed-torque curve of the motor; and 2) the limited power available from the variable frequency motor-generator set. These difficulties were remedied by increasing the diametral clearance between the motor and the stator to 0.030 in. by removal of metal from the rotor surface.

Problems associated with shaft-impeller running clearances developed from inadequate mechanical clearance between the rotary and stationary elements. The clearance problems were resolved by providing larger running clearances in the labyrinth seal area and between the inner race of the bearings and the inside diameters of matching grease seals.

Subsequent to cold shakedown, the compressor was installed in the hot test loop for mechanical and instrumentation shakedown. Initial tests were conducted at 6000 rpm and at some intermediate helium gas temperature and pressure. Later, the compressor speed was incrementally increased to 12 000 rpm and to the design operating temperature and pressure of 600°F and 400 psia, respectively. The power to drive the compressor was supplied by a 400-cycle, gasoline-driven motor-generator set.

During the hot shakedown tests, no mechanical malfunction of the compressor was experienced. Some maintenance to minimize helium leakage to the atmosphere and to make instrumentation corrections was required.

Compressor performance data at several speeds were obtained with helium at suction conditions of 400 psia and 600°F. The head-flow characteristics are shown on Fig. 3. It should be noted that these characteristics are not representative of those that would be expected for the compressor as originally designed, since it was decided to omit the straightening vanes originally planned to be installed in the vertical annulus, to permit earlier initiation of hot testing. Furthermore, the design of the flow path at the lower end of the compressor vessel was considerably less sophisticated than that originally planned for the final installation. It seems probable that the incorporation of these additional features into the hot test setup might have permitted obtaining a somewhat higher head at any given flow and speed.

It is believed that the peculiar vertical dip in the characteristic curves was caused by stalling of the diffuser vanes at low values of flow. It has been suggested that this dip could have been avoided by increasing the inside (and consequently the outside) diameter of the vanes. Such an increase was not possible within the original physical design limitations, but could be used in possible future applications.

Coolant flow tests were performed by varying the water flow and water temperature and noting the effect on bearing temperature. Because of the system gas flow geometry, a minimum water flow of 4 gpm was required to maintain a satisfactory bearing temperature. With complete loss of coolant water at design operating conditions, the bearing temperature increased from the nominal 125°F to the alarm setpoint temperature of 190°F in approximately 12 minutes.

After completion of compressor performance data-taking and various tests, an endurance run was made to determine the reliability and longevity of grease-lubricated ball bearings. The compressor was operated at 12 000 rpm with helium at 600°F and 400 psia suction conditions.

Results

The grease-lubricated centrifugal compressor completed an endurance run of 3500 hr at 12 000 rpm with helium at design conditions. The total operating time was accumulated without shutdown, except for repairs to the gasoline-driven motor-generator set. The termination of the endurance run was scheduled.

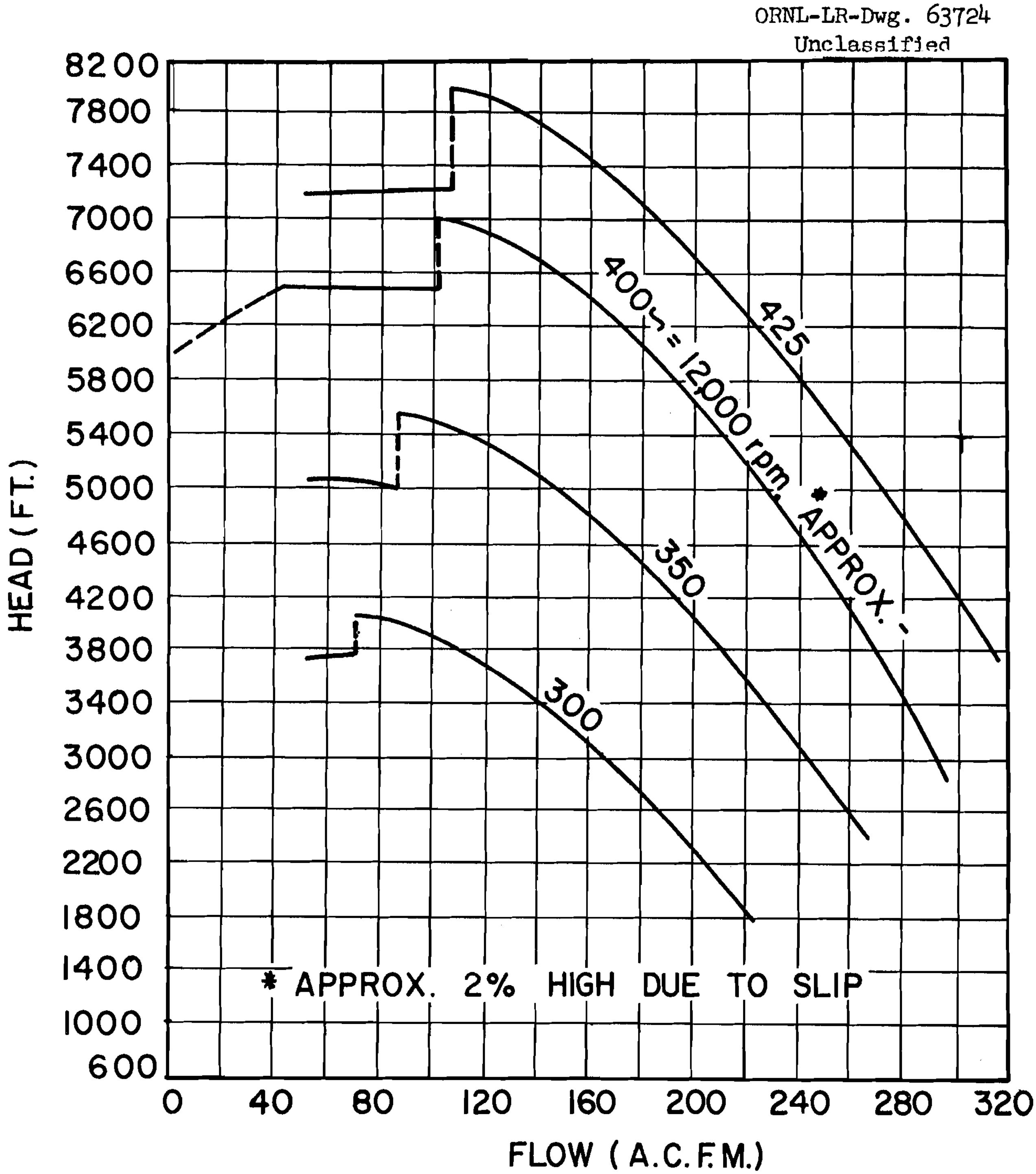

Fig. 3. Head vs Flow, HECC-1 Compressor, Helium 400 psig and 600°F

A post-test inspection was made on the compressor with particular emphasis on the ball bearings and their respective housings. The rear bearing (motor end) and grease lubricant were in good condition. The total weight loss of grease in the bearings and grease seals was on the order of 0.5 g/1000 hr of operation. The front bearing was in satisfactory condition; however, the grease had become discolored and its lubricity had deteriorated. An examination of the balls and bearings raceways indicated a normal running band with no evidence of spalling or undue wear. The deterioration of the front-bearing grease lubricant was hastened by an ineffective heat sink and closeness to the 600°F suction. Later tests showed that the cold finger assembly inserted into the hollow shaft was not circulating water coolant. Measurements taken of the front bearing housing indicated some dimensional increase and a slight taper.

Conclusions

The experience gained through the operation of a grease-lubricated centrifugal compressor indicates that reliable maintenance-free operation can be attained for at least 3500 hr when circulating helium at 600°F.

A number of compressors using the same motor, cooling, and bearing assembly as the unit described in this report, but equipped with impellers of the regenerative type, have been built and operated during the last two years. While the design and development of the regenerative compressors will be reported separately, three items are worthy of inclusion in the present report. First, it has been found that these compressors can be consistently operated for periods of at least 3000 hr without maintenance, and the inspection of bearings removed after such runs indicates that it would not be unreasonable to expect satisfactory service for periods of 5000 hr or more. Secondly, if high reliability is required, it has been found advisable to give these compressors a preliminary hot test for about 150 hr to eliminate "infant mortality". Thirdly, it has been found that the same general type of problems with the water-cooling system described for the centrifugal unit have been encountered with the regenerative units. It has been possible to cure these difficulties by Kanigen (electroless-nickel) plating the insides of the water passages. If a substantial additional number of compressors of this general type was required, however, it would seem advisable to redesign the water-cooling system to minimize such difficulties and to reduce the overall cost of the units.

In all other respects the centrifugal compressor described in this report has been a satisfactory unit which has met the original design goals.

Acknowledgments

Important contributions to the design and development of the compressor were made by several individuals. The author wishes to acknowledge W. G. Cobb and L. V. Wilson for the mechanical design, W. F. Boudreau and A. G. Grindell for formulating the developmental techniques, and D. E. Gladow for assistance in compressor testing.

References

1. Barden Precision Ball Bearings, (Series 107 H), Barden Company, Danbury, Conn.

2. Communication with Barden Company Engineers.

3. A. J. Stepanoff, Centrifugal and Axial Flow Pumps, (Wiley, New York, 1957), 2nd ed.

Internal Distribution

1. S. E. Beall
2. M. Bender
3. A. L. Boch
4. W. F. Boudreau
5. R. B. Briggs
6. D. W. Cardwell
7. W. G. Cobb
8. J. A. Conlin
9. A. P. Fraas
10. J. K. Franzreb
11. C. H. Gabbard
12. D. L. Gray
13. A. G. Grindell
14. R. B. Korsmeyer
15. M. I. Lundin
16. R. N. Lyon
17. H. G. MacPherson
18-38. W. D. Manly
39. A. J. Miller
40-45. I. K. Namba
46. A. M. Perry
47. M. W. Rosenthal
48. H. W. Savage
49. A. W. Savolainen
50. M. J. Skinner
51. P. G. Smith
52. I. Spiewak
53. E. Storto
54. J. A. Swartout
55. E. R. Taylor
56. D. B. Trauger
57. L. V. Wilson
58. C. E. Winters
59. H. C. Young
60. J. Zasler
61-63. Central Research Library (CRL)
64-66. Y-12 Document Reference Section (DRS)
67-68. Laboratory Records Department (LRD)
69. Laboratory Records Department, Record Copy (LRD-RC)

External Distribution

70-84. Division of Technical Information Extension (DTIE)
85. Research and Development Division, ORO
86-87. Reactor Division, ORO