PATHFINDER ATOMIC POWER PLANT
CONCEPTUAL DESIGN STUDY OF CRBR
QUICK-OPERATING PRESSURE VESSEL
CLOSURE AND SELF-ENERGIZING SEAL TESTS

Final Report

By
R. G. Michel

October 15, 1959

Atomic Energy Division
Allis-Chalmers Manufacturing Company
Milwaukee, Wisconsin

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FINAL REPORT
CONCEPTUAL DESIGN STUDY OF CRBR
QUICK-OPERATING PRESSURE VESSEL CLOSURE
and
SELF-ENERGIZING SEAL TESTS

by
R. G. Michel

Submitted to

U. S. ATOMIC ENERGY COMMISSION,
NORTHERN STATES POWER COMPANY

and

CENTRAL UTILITIES ATOMIC POWER ASSOCIATES

Under

Agreement dated 2nd Day of May 1957 and Supplemental
Agreement dated 22rd Day of November 1957 between
NSP and A-C under AEC Contract No. AT (11-1)-589

by

ALLIS-CHALMERS MANUFACTURING COMPANY
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MILWAUKEE 1, WISCONSIN

October 15, 1959

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FOREWORD

This report describes the work conducted by Allis-Chalmers Manufacturing Company and its subcontractors to determine the feasibility of using a quick-operating closure for the pressure vessel of the Controlled Recirculation Boiling Reactor (CRBR). Initially effort was directed toward determining the feasibility of using a segmented ring-type closure. Allis-Chalmers also recognized that development of a self-energizing seal would greatly facilitate the development of a quick-operating closure, and evaluation tests of various seal materials and designs were initiated.

It became obvious early in the program that closure designs of the segmented ring-type tended to lose their inherent simplicity when adapted for use with the CRBR. Allis-Chalmers therefore retained the services of two consulting firms -- Allstates Design and Development Company, Inc. and Koerper Engineering Associates, Inc. These firms made independent conceptual design studies to determine a feasible type of closure.

It also became obvious early in the program that the feasibility of the segmented ring-type closure was contingent on the selection of a suitable self-energizing seal. Effort at Allis-Chalmers was therefore concentrated on the seal evaluation tests.

An outline of the work performed and the conclusions made is given in Part I, Summary. Since the study of the segmented ring-type closure was foreshortened, the results are also included in the Summary.

Detailed results of the seal evaluation tests are given in Part II, Self-Energizing Seal Tests. The final report submitted by Allstates Design and Development Company, Inc. and Koerper Engineering Associates, Inc. are given as Appendix A and B, respectively.

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Part I.

SUMMARY

1.1. Introduction

Conceptual design studies of quick-operating closures were made to determine a type suitable for the pressure vessel of the Controlled Recirculation Boiling Reactor (CRBR). The design requirements of the closure are as follows:

- 1) The closure should be capable of being opened or closed in 30 minutes or less.
- 2) The closure should be capable of containing saturated steam at 500 F and 600 psig.
- 3) A closure clearance opening of 7 feet 3 inches was required. (The present requirement is 7 feet 7 inches.)
- 4) The closure and locking mechanisms are to be suitably insulated and sealed from contact with the shield pool water above the reactor vessel.
- 5) The closure locking and sealing devices should be capable of withstanding the design pressure and temperature for a period of about four months, and should be such that the closure may be readily opened thereafter. The design life of the vessel is twenty years, and the closure would be opened three or four times each year.
- 6) A leakage rate of 5 pounds per day from the primary vessel seal would be acceptable if a secondary seal is used. Leakage is to be collected through a suitable drainage system. Erosion or cutting

of the gasket surfaces by such leakage is not permitted.

7) Heavy metal sections in the closure region are to be avoided since the pressure vessel must be capable of being heated to operating temperature or cooled down to shutdown temperature in about an hour without inducing excessive thermal stresses.

8) The closure design should include a suitable indexing device so that the control rod drives mounted on the cover may be properly positioned with respect to the control-rod guide tubes located in the pressure vessel.

9) The closure should be designed so that the closure may be opened or closed under 18 feet of shield pool water, i.e. remotely.

1.2. Conceptual Design Study

1.2.1. Segmented Ring-Type Closure

A segmented ring-type closure, a basic type of closure that is used in the field of high pressure feedwater heaters was investigated by Allis-Chalmers Manufacturing Company. One of the completed conceptual designs is shown in Figure 1.1. The closure incorporates a double seal consisting of the primary, flexible, self-energizing seal ring and a back-up seal of metallic rings. Differential movements between the head flange and vessel flange during pressurizing is taken up in the torus ring seal. Two seals are used to prevent the shield pool water from coming into contact with the hot vessel surface during operation.

1.2.2. Conceptual Designs -- Allstates Design and Development Company, Inc.

The design concepts that Allstates Design and Development Company, Inc.

recommend (Appendix A) are of the types using conventional compression gaskets. The closures are designed so that bolting large enough to apply only the necessary gasket loading is required, and operating pressure forces on the vessel head are retained by other means, e.g. large threads, shear blocks, C-clamps, etc. Rough calculations and preliminary investigations indicate that such locking devices are feasible.

1.2.3. Conceptual Designs -- Koerper Engineering Associates, Inc.

Although the quick-operating closure designs investigated by Koerper Engineering Associates, Inc. include mechanisms using both compression and self-energizing seals (Appendix B) the primary emphasis is on those designs using self-energizing seals. The investigation of several seal designs and materials is described, but no entirely suitable material or design is indicated. The report indicates, however, that a suitable material for a self-energizing seal is probably now available but considerable data must be obtained before a suitable seal can be designed.

1.3. Self-Energizing Seal Tests

Tests were conducted to determine a material and design that would be suitable for use as a self-energizing seal for a quick-operating closure for the CRBR. Details of these tests are given in Part II. Results indicate that either virgin Teflon or asbestos-impregnated Teflon of an E-shaped design may be feasible. Both materials exhibit a high shrinkage rate, however, and further data on this characteristic is necessary before a Teflon seal can be reliably designed. In addition,

Teflon is one of the plastic materials most readily damaged by irradiation (ref 1,2,3). Calculations show that the gamma irradiation present at the closure will break down the material in an hour or less. Adequate shielding would therefore have to be developed if this material is used.

1.4. Conclusion

A quick-operating closure for the CRBR pressure vessel is certainly desirable, and the study describes a number of designs using both conventional compression seals and self-energizing elastic seals that may be feasible. All closures proposed, however, tend to become complex in design, operation, or manufacture. The complexities arise primarily because the closure must not only contain the internal pressure of the vessel but must also prevent shield pool water from coming in contact with the vessel head.

In addition to these complexities, extensive evaluation programs would be necessary before any one of the proposed closures could be considered entirely satisfactory for full-scale testing. A closure using conventional compression seals, for example, would require large, threaded, locking rings or sliding-wedge locks; and a low friction, non-galling material combination would be required. Such a material combination would have to be determined through an evaluation program. In addition, devices would have to be developed with which the closure may be remotely positioned and locked in place without compromising the quick-operating features and without causing significant distortion of the closure.

If a self-energizing elastic seal capable of withstanding the reactor environment is developed, a closure that is easier to design, manufacture and operate can be used. The study indicates that such a material may be available at present, but an extensive evaluation program is required to find the material.

The amount of data thus indicated to be necessary, before development of a particular design can be justified, cannot be easily included in the CRBR program; and the success of a concentrated development effort without benefit of such data would be unpredictable. The program to develop a quick-operating closure for the CRBR pressure vessel is therefore not recommended.

Part 2.

SELF-ENERGIZING SEAL TESTS

2.1. Introduction

A test program was initiated to evaluate seal designs and materials for possible use in a quick-operating closure of the pressure vessel of the Controlled Recirculation Boiling Reactor (CRBR). Design requirements of a suitable seal are as follows:

Design Pressure	660 psig
Design Temperature	500 F
Design Life	4 months (minimum)
Leakage Rate	5 pounds/day (maximum)

The designs and materials considered for the tests were based on manufacturers' recommendations and included the following:

Silicone rubber compounds

JM #809 - Johns Manville - E-Shaped

K-1035 - Union Carbide - C-Shaped

S-2098U - Dow Corning - C-Shaped

Polytetrafluorethylene

Virgin Teflon - E.I. du Pont de Nemours Company

E-Shaped

Asbestos-Filled Teflon - E.I. du Pont de Nemours Company

E-Shaped

Fluorelastomer

Viton A - E.I. du Pont de Nemours Company - E-Shaped

Carbon Steel

SAE 1010 - Wedge-shaped

2.2. Test Equipment and Procedures

The autoclave designed for the test is shown in Figures 2.1 and 2.2. A segmented ring-type closure (Figure 2.3) that accommodates E-type, C-type, and metallic wedge-type seals with outside diameters of 12 inches is used. The closure is designed so that it can be easily modified to accommodate other types of seals.

Leakage from the seal is conducted from above the closure through a condensing line to a collection beaker containing oil. A 15 kw immersion heater that is controlled by an on-off power switch regulated by thermocouple heats the water to the desired temperature. A pressure switch is also included in the immersion heater circuit to prevent excessive pressure. A rupture disk is provided for safety. A liquid-level indicator is not used since the leakage is collected, and the amount of liquid in the autoclave can be accurately determined at any time.

A volume of water is measured at room temperature to establish the water level at the start of each test. The seals are tested for four hours under the following conditions: 900 psig (hydrostatic pressure) - room temperature; and at saturated steam pressures of 100, 200, 300, 400, and 500, psig. The amount of leakage is visually observed and the, and the volume is measured. After each 4-hr exposure, the test is stopped, and

the seal is examined. If the material passes these tests, exposures of 90 days at reactor operating temperature and pressure are made.

2.3. Results

2.3.1. Test Equipment

The test equipment operated properly throughout the tests with the following exceptions:

- 1) The heater flange gasket, flat pressed asbestos (J-M Cranite) failed on two occasions on resumption of tests, shortly after a shutdown. The gasket deteriorated while drying out during the period between tests. The gaskets were therefore replaced after each shutdown.
- 2) The rupture disk failed in two instances during normal operation. The use of the usual pop-off safety valve between the autoclave and rupture disk would probably have precluded these failures.

2.3.2. Preliminary Tests of Materials and Designs

E-Shaped Silicone Rubber Compound (J-M #809) -- The material was exposed in air in an electric furnace at 500 F for 4 hours. The material was slightly harder and the dimensions were slightly changed by the exposure.

The material was then placed in the test closure and given 4-hour exposures at successively higher temperatures. The seal failed at about 475 F and 455 psig. The failed seal was bloated and tacky.

C-Shaped Silicone Rubber Seals (K-1035 and SI086U) -- The manufacturer had experienced molding difficulties, and the dimensions of the seals received were not accurate. The seals failed in the hydrostatic test.

In view of the previous failure of a silicone rubber compound, no further tests of this material were made.

E-Shaped Virgin Teflon -- The seal energized at the beginning of the test and sealed the closure properly at 600 psig. When the seal was examined after the test, however, shrinkage was observed.

The seal was used in a second test, but failed to energize on application of pressure until it was preheated. The seal performed properly otherwise.

The seal was used in a third test and again required preheating before it would energize. The seal visibly leaked at lower pressures but the leakage decreased when a pressure of 250 psig was reached. The seal continued to visibly leak until a pressure of 500 psig was reached. Although leakage could not be then observed, a leakage rate of 4 cubic centimeters of water per week was measured. This leakage continued for about 4 weeks and then stopped. The test was run for 70 days and was terminated because the pressure regulator failed. No visible change in the material was observed at the completion of the test.

E-Shaped Asbestos Impregnated Teflon Seals -- The seal was tested by manually regulating the pressure. The seal energized immediately, and was exposed for 3 hours without visibly being affected. The seal would not energize in successive tests unless the seal was heated before applying pressure.

E-Shaped Viton A -- A sample of Viton A was suspended in the autoclave for 3 hours under reactor operating conditions, and was then

examined. The material was blistered, and the dimensions had increased by 25 per cent. Since the results were unsatisfactory, no further tests were made.

Wedge-Shaped Metallic Seals -- In view of the discontinuation of the quick closure conceptual design study this seal was not evaluated.

2.3.3. Shrinkage Characteristics of Teflon

Introduction -- Since Teflon seals failed to energize after initial exposure to reactor conditions, apparently because of shrinkage, tests were made to determine the rate of shrinkage. Four samples, three of which had been previously exposed to high temperatures and pressures, were placed in an autoclave and exposed to steam and water at 500 F and 660 psig for 70 days. The dimensions of the samples (Figure 2.4) were taken before and after the tests.

Results -- Detailed results of the tests are given in Table 2.1. A photograph of an exposed sample is given in Figure 2.5. The discoloration of the sample is unusual, since the sample had been exposed to the same conditions as were used in the final exposure. Two possible explanations of this discoloration are 1) the presence of impurities in the water, and 2) deterioration of the sample due to exposure to temperatures above 500 F. The water was changed after each test, and impurities may have been introduced in the final test. During a previous test, the sample was heated above 500 F when the controls failed, and a marked increase in shrinkage was noted in the sample at that time.

TABLE 2.1

DIMENSIONS
OF
TEFLON SEALS AND SAMPLES
AFTER EXPOSURE TO SATURATED STEAM AT 600 PSIG

DIMENSION* CODE	SAMPLE**	DIMENSION IN INCHES AFTER				
		0 Hours	6 Hours	11 Days	17 Days	70 Days
A	1	0.732 IN.				0.685 IN.
	2	0.732 IN.				0.675 IN.
	4	0.732 IN.	0.745 IN.			
B	1	0.850 IN.				0.767 IN.
	2	0.850 IN.				0.780 IN.
	4	0.850 IN.	0.770 IN.			
C	1	0.800 IN.				0.935 IN.
	2	0.789 IN.				0.985 IN.
	3	0.800 IN.				0.787 IN.
	4	0.800 IN.				0.786 IN.
D	3	10.50 IN.				10.34 IN.
	4	10.50 IN.	10.42 IN.			
E	1	3.388 IN.		3.340 IN.	3.332 IN.	3.266 IN.
	2	3.509 IN.		3.495 IN.	3.487 IN.	3.178 IN.

* The dimension code is given in Figure 2.4.

** The sample code is as follows:

- 1) New asbestos-impregnated Teflon sample.
- 2) Regular Teflon sample cut from seal previously exposed for four hours.
- 3) New Teflon seal.
- 4) New asbestos-impregnated Teflon seal.

2.4 Conclusions

Of the materials studied, only Teflon appeared suitable for use as a self-energizing seal when exposed to operating temperature and pressure for prolonged periods. A shortcoming of the material for this application is that the seal failed to energize after initial use unless the seal was first heated. This problem was apparently caused by shrinkage of the material. Tests show the shrinkage rate of Teflon to be very rapid during the initial exposure and to be slower after this exposure. This indicates that it may be possible to temper the material before manufacturing the seal and thereby reduce the shrinkage to a tolerable rate.

Another shortcoming of Teflon is the irradiation damage to which it is subject. Calculations indicate that gamma irradiation about the reactor closure would make the material unserviceable in an hour or less.

References

1. D. S. Billington, Radiation Damage in Reactor Materials, ("Proceedings of the International Conference on the Peaceful Uses of Atomic Energy.") Vol. VII, P/744, United Nations, 1955.
2. Effects of Irradiation on Packing and Gasket Materials, Johns-Manville Report DP-26-6, April 30, 1957.
3. D. L. Peterson, Preliminary Rubber Irradiation Tests, AEC Report HDC-1876, August 25, 1958.

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REACTOR VESSEL
COVER SCHEME
MODEL "B"

REACTOR VESSEL CLOSURE SCHEME, MODEL B FIG. 1.1

SIMILAR TO: 43-001-888 SCALE 3 = 12
SHEET /
OF / **43-201-589**

43-400-824

REQ	ISS	DEL	WHS	QTY	UNIT	ITEM	DESCRIPTION	MATERIAL	PART NUMBER	✓
1	1	1	1	1	1	1	SEAL TEST ASSEMBLY		33-400-824	
1	1	1	1	1	1	2	1/2" DIA. 1/8" WALLS PIPE	ASTM A182	THIS	0001
1	1	1	1	1	1	3	FLANGE 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0002
1	1	1	1	1	1	4	BOLT 1/2" DIA. x 1/8" x 1/8" DIA. PLATE	ASTM A182	THIS	0003
1	1	1	1	1	1	5	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0004
1	1	1	1	1	1	6	COVER W/ PLATE	ASTM A182	THIS	0005
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1	1	1	1	1	1	69	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0068
1	1	1	1	1	1	70	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0069
1	1	1	1	1	1	71	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0070
1	1	1	1	1	1	72	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0071
1	1	1	1	1	1	73	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0072
1	1	1	1	1	1	74	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0073
1	1	1	1	1	1	75	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0074
1	1	1	1	1	1	76	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0075
1	1	1	1	1	1	77	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0076
1	1	1	1	1	1	78	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0077
1	1	1	1	1	1	79	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0078
1	1	1	1	1	1	80	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0079
1	1	1	1	1	1	81	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0080
1	1	1	1	1	1	82	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0081
1	1	1	1	1	1	83	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0082
1	1	1	1	1	1	84	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0083
1	1	1	1	1	1	85	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0084
1	1	1	1	1	1	86	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0085
1	1	1	1	1	1	87	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0086
1	1	1	1	1	1	88	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0087
1	1	1	1	1	1	89	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0088
1	1	1	1	1	1	90	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0089
1	1	1	1	1	1	91	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0090
1	1	1	1	1	1	92	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0091
1	1	1	1	1	1	93	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0092
1	1	1	1	1	1	94	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0093
1	1	1	1	1	1	95	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0094
1	1	1	1	1	1	96	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0095
1	1	1	1	1	1	97	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0096
1	1	1	1	1	1	98	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0097
1	1	1	1	1	1	99	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0098
1	1	1	1	1	1	100	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0099
1	1	1	1	1	1	101	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0100
1	1	1	1	1	1	102	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0101
1	1	1	1	1	1	103	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0102
1	1	1	1	1	1	104	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0103
1	1	1	1	1	1	105	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0104
1	1	1	1	1	1	106	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0105
1	1	1	1	1	1	107	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0106
1	1	1	1	1	1	108	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0107
1	1	1	1	1	1	109	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0108
1	1	1	1	1	1	110	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0109
1	1	1	1	1	1	111	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0110
1	1	1	1	1	1	112	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0111
1	1	1	1	1	1	113	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0112
1	1	1	1	1	1	114	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0113
1	1	1	1	1	1	115	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0114
1	1	1	1	1	1	116	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0115
1	1	1	1	1	1	117	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0116
1	1	1	1	1	1	118	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0117
1	1	1	1	1	1	119	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0118
1	1	1	1	1	1	120	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0119
1	1	1	1	1	1	121	WELD 1/2" x 1/8" x 1/8" PLATE	ASTM A182	THIS	0120
1	1	1	1	1	1

NOTE #2:
FINISH ALL OVER UNLESS OTHER-
WISE SPECIFIED - BREAK ALL SHARP
EDGES

NOTE #3:
ASSEMBLED TESTER TO BE HYDROSTATIC
TESTED ACCORDING TO TEST. SPEC. 43-T-1311

NOTE #4:
FLASH NICKEL PLATE (0005 MAXIMUM
THICKNESS) THE OUTSIDE DIA. OF IT 16 AND
THE 10.425 X 1.002 TURN ON IT 5

SCAL TEST ASSEMBLY		1-1/2" x 1-1/2" x 1-1/2"		43-400-824																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																
<table border="1"> <tr> <th colspan="2">ITEM NO.</th> <th colspan="2">QUANTITY</th> <th colspan="2">REMARKS</th> </tr> <tr> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>2</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>3</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>4</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>5</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>6</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>7</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>8</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>9</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>10</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>11</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td>12</td> <td>1</td> 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ITEM NO.		QUANTITY		REMARKS																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																
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SEAL TEST ASSEMBLY FIG. 2.1

SEAL TESTER WITH SILICONE RUBBER E-SEAL IN PLACE

FIG. 2.2

SILICONE RUBBER E-SEAL AND METALLIC WEDGE-TYPE
SEAL WITH COVER AND SEGMENTED RING

FIG. 2.3

FIG. 2.4 – DIMENSION OF TEFLON SAMPLES

FIG. 2.5 – TEFLON SAMPLE AFTER 70-DAY EXPOSURE. THE TOP SAMPLE IS NEW. THE BOTTOM SAMPLE WAS EXPOSED TO SATURATED STEAM AT 600 PSIG FOR 70 DAYS.

Appendix A

CONCEPTUAL DESIGN STUDY OF A QUICK OPENING CRBR PRESSURE VESSEL CLOSURE

This report covers work performed under Allis-Chalmers Manufacturing Company's Purchase Order No. WA-440304-NSP, a part of the Research and Development Program under AEC Contract No. AT(11-1)-589 with Northern States Power Company.

Allstates Design and Development Co., Inc.
322 East Michigan Street
Milwaukee 2, Wisconsin

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Original Sketches and "Talking papers"

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SUMMARY

A conceptual design study of a quick acting closure for a CRBR Reactor Pressure Vessel is presented. Methods of closure locking and retention are discussed. General closure configurations were investigated. Closure locating and registering methods are considered. The heart of the problem, the gasketing or sealing design, was investigated and several possibilities are presented along with the advantages and disadvantages of each. An extensive bibliography is included in the appendix, as are several publications. Conclusions and recommendations for avenues of further feasibility study are indicated.

INTRODUCTION

The basis for this study is the "Design Specifications for the CRBR Reactor Pressure Vessel Closure" (see Appendix) in which are given the primary limitations and design requirements.

It was found that very little information on closures of this size is available, the best sources of information being the design and data of operating reactor pressure vessels such as heat exchangers, and some techniques used in the petroleum industry. (paragraph on seals)

This study is largely limited to the consideration of closure locking and retention and gasket and sealing means. It early became obvious that the traditional flange and stud arrangements have become so standard that little or no experience is available on other methods of closure locking.

The means of remote, underwater handling of various elements and the closure itself have been considered and mentioned but are not the subject of this study.

It will be noted that much of this report is presented in an outline form since it is felt that no useful purpose is served by long involved prose. To this end copious use has been made of tabulations and short factual statements.

CONCLUSIONS AND RECOMMENDATIONS

Since the object of this study is of a conceptual nature, the limited time was spent on arrangements rather than on quantitative analysis other than rough calculations to arrive at approximate sizes and numbers. It is felt that the recommendations will indicate qualitative conclusions in an effort to narrow the number of designs feasible enough to warrant further vigorous analysis.

GENERAL CLOSURE CONFIGURATION

It became obvious that a hemispherical closure will produce the least distortion of the flanges, especially if a 10" to 12" cylindrical section is interposed between the hemispherical portion and the closure flange. In view of the lack of evidence that gaskets with appreciable "spring back" can be designed, it is deemed imperative that flange "rotation" and relative (vessel expansion due to temperature and pressure is of the order of 5/32" on the diameter) movement be eliminated insofar as is possible. The general configuration of page 14 is recommended.

CLOSURE RETENTION

Of all the concepts presented it was concluded that the following appear to present the best possibilities for vigorous design investigation.

1. An arrangement based on the use of a large diameter, acme-type thread is one of the best possibilities for achieving the 30 minute disassembly time factor. cursory investigation has indicated that it is possible to machine such a thread. Such a design may be arranged either with or without bayonet "fingers" depending on the developing of a feasible means of starting a large diameter thread. The use of "fingers" allows the threaded members to remain permanently engaged.

Except for the thread, all parts are relatively simple to machine and cladding and corrosion prevention is relatively simple. The arrangement is adaptable to any of the recommended types of gaskets. Its disadvantages are the accurate machining required to produce a suitable thread and the problems of designing a suitable driving mechanism.

See sketch pages 5, 6 and 7.

2. An arrangement making use of an external segmented retaining ring designed to that sectors swing outward from the flanges. Sketch 37. This scheme also has a good possibility of meeting the assembly time requirements, but has the disadvantage of requiring the use of screws (which can be captivated) for gasket compression and closure retention.

Advantages of this style are:

- Gasket compression forces can be applied directly over the gasket minimizing the effects of flange distortion.
- Simple machining with narrow flanges.
- All parts can be captivated.
- Although some 50-60 2-3/4" screws will be required, the torque to turn them is limited to the force required for gasket seating.
- A relatively simple screw driving mechanism can be used.

3. A design using studs and/or screws located external of the flanges and stabilized clamping rings. A choice can be made as to whether the rings are segmented and swing away from the flanges or a bayonet type of "finger" is applied to the lower clamping ring. Either arrangement will allow for quick disassembly after loosening the studs and/or nuts.

The main advantages of this design are:

- Very narrow flanges may be employed and retention forces are very nearly above the gasket location.
- Machining and tolerance problems are not severe and cladding is relatively simple.
- Any of the recommended gasket types can be incorporated and all parts can be captivated.
- A straightforward stud driving mechanism may be used but the time consumed by loosening and tightening 50-60 studs is somewhat of a disadvantage. See sketch page 8.

4. A variation of the conventional stud design using very long sleeves of a material, like 18-8 stainless steel, having a materially greater coefficient of expansion than the stud material. This scheme has the advantage of requiring studs to be tightened only enough for initial gasket seating, as the vessel and closure are heated additional force is available from the expanding sleeves to retain the closure and seat the gasket farther. After cooling the studs would be relatively easy to loosen. All the advantages of (2) and (3) are available in this design except (like (3)) that it is impossible to apply gasket compression forces directly to the gasket center.

See sketch, page 9 and 10.

THREADED WITH BAYONET & FLAT GASKETS

5

SKETCH NO. 1

"MASON JAR" THREADED
WITH FLAT GASKETS

SKETCH NO. 2

THREADED WITH SELF
ENERGIZED GASKET

7

A11

ST'L COEF 6.6 (10⁻⁶)

SKETCH NO. 5

STUD WITH EXPANSION
SLEEVE & DOUBLE
GASKET.

A13

GASKETS

Two types of gaskets seem to be best suited and are recommended:

1. A flat gasket (if suitable handling means can be devised, it can be profiled) of soft copper or aluminum plated with .003 to .005 of silver. This combination has many advantages:
 - It can be designed with thick, rugged sections.
 - It does not require fine finishes on the mating surfaces and the mating surfaces can be flat and easy to machine.
 - Leakage drain off arrangements are simple to achieve.
 - Gasket seating forces are only slightly higher than other configurations. See sketch, pages 55 and 69.
2. A triangular gasket of soft steel also silver plated (see sketch page 53) and applied with a retention design so arranged that, like sketch on page 25, the hydrostatic forces on the closure are used to seat the gasket. This arrangement has been successfully used in smaller sizes and has most of the advantages of the flat, solid type gaskets. Assembly difficulties may be encountered, since the gasket is a part of the closure.

GENERAL CLOSURE CONFIGURATIONS

Shapes for this closure fall into several basic types, namely flat, hemispherical, ellipsoidal, tairspherical (spherically dished).

The flat closure has many difficulties. Among them are:

1. Heavy thickness required.
2. Bulging (especially when thinner sections are used) produces extremely high local stresses on studs, nuts or other retaining devices.
3. Expansion of vessel under pressure tends to produce relative movement between the closure and the vessel, presents an almost impossible gasket condition.

The flat closure however does have some virtue, it is simple to fabricate and machine, accommodates control and mechanisms nicely and its use reduces overall height of the vessel.

A flat cover could be applied to the top of a cylinder extending upward from the closure flanges but although easy to fabricate this arrangement would have all the above disadvantages to some degree.

The hemispherical shaped closure although requiring a relatively thin section is difficult to fabricate but has many advantages.

1. Hydrastatic forces do not cause large distortions and only the hoop stresses are of consequence.
2. Temperature and pressure cause expansions at about the same rate and direction as in a cylinder, producing favorable gasket conditions.

There are the disadvantages of increased height, difficult penetrations for control rod and other mechanisms, and less space available for manipulation of the closure locking arrangements.

The evils of the flat plate and the difficulties of the hemispherical type can both be alleviated to some degree by use of either the ellipsoidal or tairspherical configurations. The amount depending on the design.

All these configurations can be improved by the introduction of a cylinder 10" to 12" long between the "knuckle" and the closure flanges.

FLAT

t = THICKNESS OF COVER IN"

d = DIA. SHORTEST SPAN IN"

S = MAX. ALLOWABLE STRESS PSI

C = .162 FOR PLATES RIGIDLY BOLTED

P = DESIGN PRESSURE PSI

$$t = d \sqrt{\frac{CP}{S}}$$

$$t = 87" \sqrt{\frac{.162 \cdot 660}{17,500}}$$

$$t = 7\frac{3}{8} \text{ AX.}$$

SKETCH NO. 8

HEMISPHERICAL

t = THICKNESS IN.

P = DESIGN PRESSURE PSI

S = ALLOWABLE STRESS PSI

D = INTERNAL DIA.

$$t = \frac{PD}{4S}$$

$$t = \frac{660 \cdot 87}{4 \cdot 17,500}$$

$$t = \frac{13}{16} \text{ Ax.}$$

SKETCH NO. 9

ELLIPSOIDAL

t = THICKNESS IN

P = DESIGN PRESSURE PSI

D = INSIDE DIA.

S = MAX. ALLOWABLE STRESS.

K = HEAD PROPORTION FACTOR $D/2h$.

$D/2h$ = RATIO MAJOR TO MINOR AXIS

h = $1/2$ LG'TH OF MINOR AXIS.

E = LOWEST EFF. OF ANY JOINT ASSUME 1

$$t = \frac{PDK}{2SE - 0.2P}$$

$$t = \frac{660 \cdot 87 \cdot 76}{2 \cdot 17,500 \cdot 1 - 0.2 \cdot 660}$$

$$t = 1 \frac{13}{16} \text{ Av.}$$

SKETCH NO. 10

TORISPHERICAL

E = LOWEST EFF. ANY JOINT ASSUME 1

t = THICKNESS IN.

P = DESIGN PRESSURE 660 PSI

L = CROWN RAD. IN. 87"

r = KNUCKLE RAD. IN. 10"

M = RATIO OF HD. PROPORTION L/r

S = ALLOWABLE STRESS PSI

$$t = \frac{PLM}{2SE - 0.2P}$$

$$t = \frac{660 \cdot 87 \cdot 1.48}{2 \cdot 17500 \cdot 1 - 0.2 \cdot 660}$$

$$t = 2\frac{1}{2}$$

SKETCH NO. 11

CLOSURE RETENTION AND LOCKING

The main considerations for the retention and locking of the closure to the vessel are:

- (1) The closure must be able to be assembled or removed in 30 minutes if possible.
- (2) The openings in the closure must register with the mating parts inside the vessel within plus or minus $1/32''$. This condition practically dictates that closure be removed and assembled in a vertical plane without rotation.
- (3) All operations connected with closure assembly and removal must be performed remotely and under water, indicating the retention and locking means must be simple, and easy to operate.
- (4) Because of its size and the difficulty of handling parts of this diameter, machining operations must be kept to a minimum and confined to turning and milling as much as possible.
- (5) The thickness of sections should be as small as is consistent with the stresses involved to minimize the thermal shock effects during rapid heating and cooling (400°F in 60 minutes).
- (6) Any arrangement must be such that it cannot be inadvertently unlocked while any steam pressure remains in the vessel.

The main possibilities of closure retention and locking means generally:

- (1) The classic horizontal flanged joint retained by studs and nuts. (Flange and Stud).
- (2) A "bayonet" type of quick acting retainer and its sister the interrupted thread.
- (3) The segmented retaining ring of either an internal or external assembling arrangement.
- (4) An arrangement in which a common shear member is introduced into both the closure and the vessel.

- (5) An arrangement using a large diameter thread similar to the old fashioned "mason jar".

It is to be noted that there is an almost infinite number of combinations of these five basic mechanisms.

The classic flange and stud (Drawing # 1) joint has historical experience, practically every vessel constructed to date, has used this method. This, the simplest of all conceivable arrangements, involves the holding of the closure flange to the vessel flange by means of studs and allows the same members (studs) to apply both the locking force and gasket compression force in one operation.

Sketches 23 and 24 show an application of single and double flat gaskets, also a self energized gasket arrangement.

A variation of this arrangement may be made by using studs with annular tongues in conjunction with split retainers and keepers (similar to the arrangement of automobile engine valve springs and valve stems). See sketch 26. In the use of this scheme a separate means of gasket compression must be provided, since the split washers must be assembled to the studs with clearance and the clearances taken up after assembly.

A. Advantages

1. Simplicity of machining and fabrication.
2. Versatility, practically any type of gasketing can be applied.
3. Can be designed with relatively thin and uniform sections, minimizing thermal shock under rapid heating and/or cooling.
4. Registration of parts is relatively simple by using two of the studs as locating pins.
5. Has small possibility of closure being accidentally released while any pressure remains in the vessel.
6. With the application of a self energized type of gasket, stud tension stresses can be limited to the hydrastatic forces only.

B. Disadvantages.

1. Tends to create a compromise between a small number of studs (To facilitate assembly and disassembly) and a large number of studs to eliminate high stud stresses.
2. In flat or nearly flat closures, high unit stresses and some bending tend to be induced in the studs and nuts due to closure rotation under pressure.
3. It is not thought feasible to captivate the nuts and spacers, to eliminate a handling and storing problem during assembly and/or disassembly.

BAYONET

In the use of this method a movable member is so constructed that its movement allows its "fingers" to interfere with "fingers" of another member, thus preventing separation of the closure and vessel. Release of the closure is affected by rotating the movable member so that its "fingers" no longer interfere.

A so called "finger" or "bayonet" type of closure locking and retaining method was evolved from study of various quick-acting locking mechanisms such as the breech closing arrangement on torpedo tubes. Although Sketch # 30 shows only single rows of interlocking fingers, multiple rows (at the cost of difficult machining) may be incorporated. Also an interrupted thread arrangement can be used but at the cost of even more difficult machining problems without gaining appreciable advantage. The use of threads or inclined surfaces as means of applying gasket pressure creates very high compression stresses on the mating members, with the accompanying galling, driving and lubrication problems.

Therefor designs in this category all include a method of engaging or disengaging closure in an unstressed or loose condition with an arrangement (usually screws) to take up clearances and apply gasket pressure.

ADVANTAGES & DISADVANTAGES

1. Relatively simple to machine (some milling or shaping required).
2. Operation is simple and the parts are rugged.

3. Stresses are nominal, but in some instances, screws (for applying gasket pressure and taking up clearances) must withstand full hydrastatic forces but need be turned under gasket compression forces only.
4. Registering is difficult but not impossible in the arrangements in this category. At least one of the methods shown in sec. 70 are applicable.
5. Every type of gasket can be applied, single and double flat gaskets and self-energized types are shown in sketches 28 & 29.
6. Various sections are quite intricate necessitating careful consideration of heating and cooling rates to avoid poor thermal shock conditions.
7. By incorporating a slight keying effect (See sketch 28) at the engaging faces of the fingers, the possibility of unlocking the closure under pressure conditions is eliminated.
8. It is a disadvantage inherent in this type of lock that at best a little less than 50% of the periphery of the flanges can be engaged, creating high bending and shear stresses and requiring that flanges be thick enough to bridge the open spaces.
9. Assuming that all members are fabricated from similar materials serious thermal or pressure expansion problems are foreseen.

LOCKING RINGS

As the term implies, this general class of closure locking and retention involves the use of a ring to prevent the separation of the closure and the vessel.

Locking Rings may be either internal or external engaging types. Either type must be segmented in some manner and may engage the vessel, the closure, or both. A variation which allows the use of unsegmented locking ring is the substitution of threads for the annular tongues and grooves. However, the accurate machining required makes this arrangement rather less unattractive.

A further non-segmented variation can be affected by interrupting either the annular tongues and grooves or threads, allowing separation by rotation of the ring similar to the "fingers" of the bayonet type. It is to be noted that the American Locomotive heat exchanger closure is of the internal segmented ring type with two flat gaskets. This closure has been used successfully on some 200 applications on diameters up to 48 inches, pressures to 4700 lbs. and temperatures to 700°F. It is however, difficult to assemble, has a tremendous number of gasket compression screws and has always been used with a very thick flat head.

ADVANTAGES & DISADVANTAGES

1. Somewhat complicated and requires precise machining.
2. Operation can be very quick and simple requiring only the loosening of gasket compressing screws and pivoting or withdrawing of ring segments. Some arrangements would require remote handling and storage since captivating is not possible.
3. Stresses (See point 3 on page 20).
4. Registering (See point 4 on page 20).
5. All types of gaskets can be applied as shown in Sketches 35-37.
6. Point 6 on page 20 applies to this arrangement also.
7. By special tongue and groove design (Sketch # 37) the segmented ring parts can be self-locking and impossible to remove so long as the vessel is pressurized.
8. This type of closure lock enjoys the advantage of 100% peripheral engagement (except in the case where annulars or threads are interrupted to avoid segmenting).
9. Assuming that all parts are fabricated from similar materials serious thermal or pressure expansion problems are not present.

SHEAR MEMBER

In general this method introduces the use of a shear member, placed part in an annular groove in the vessel and part in the closure. The shear member being removable.

Inherently, the use of some auxiliary method of gasket compression is dictated since a shear member is impossible to remove under pressure. Also the arrangement is complicated by the difficulty of introducing a shear member into a necessarily practically closed annular cavity. See sketch. 42

ADVANTAGES & DISADVANTAGES

1. Machining difficult.
2. Operation simple but some parts, especially an articulated shear ring are not rugged.
3. Stresses are nominal since practically 100% of periphery can be engaged.
4. Registering can be easily accomplished in a variety of ways.
5. All types of gaskets can be applied.
6. Sections are not intricate or large minimizing thermal expansion and pressure expansion problems.
7. In general, friction of between members under pressure makes parts so difficult to move that accidental opening of closure is impossible.

SKETCH NO. 15

SPLIT STUD RETAINER
& SLEEVE KEEPER W/
DOUBLE GASKET

SKETCH NO. 16

40-3" STUDS ON 100" DIA.
 WORKING STRESS 30,000 PSI
 TOTAL FORCE 6,500,000⁺ A31

SPLIT STUD RETAINER &
THREADED KEEPER W/
DOUBLE GASKET

SKETCH NO. 17

BREECH TYPE RETAINER
WITH A SELF ENERGIZED
GASKET

SKETCH NO. 23

SEGMENTED EXTERNAL
RETAINER & LOCK-
ING RING W/
DOUBLE GASKET

SKETCH NO. 26

EXTERNAL SEGMENTED
RETAINER WITH FLAT
GASKET.

SKETCH NO. 27

INTERNAL SEGMENTED
RETAINER WITH DOUBLE
GASKET

SKETCH NO. 30

SEGMENTED RING
RETAINER WITH SELF
ENERGIZED GASKET

SKETCH NO. 31

A46

SEGMENTED RETAINING RING DETAIL

AIR CYLINDER
AX. 6" STROKE

GASKETS

A study of Gaskets is a considerable undertaking. In the course of this investigation, information was gathered from manufacturers of gaskets, data from Argonne Laboratories and various other sources.

For the sake of simplicity, most of the applicable types will be listed and diagrammed in a somewhat tabular form with some pertinent remarks on each, further details being available in the appendix and bibliography.

Gaskets generally divide themselves into two broad fields.

1. Gaskets compressed directly between flanges.
2. Gaskets compressed radially between a pilot surface on the closure and the inside periphery of the vessel.
See Page 53 for illustration.

These groups can be further characterized as:

- (1) Those achieving a seal by a compression force not exceeding the elastic limit of the gasket metal.
- (2) Those achieving a seal by exceeding the yield point of the gasket metal, and materially distorting the original gasket shape.
- (3) Those producing a seal by compressing or deforming bent or formed sheet metal usually in combination with a filler material.

In general there are two types of methods for compressing gaskets; one, obtaining the entire compressive force from the closure retaining devices and two, allowing the internal pressure to supply the compression force using the closure retaining mechanism for initial compression force only.
See Page 43.1 Fig. 172.

$$A \text{ MUST} = B + C$$

$$A \text{ MUST} = C$$

(INITIALLY
A MUST = B ONLY)

SKETCH NO. 32

For the purposes of this closure, investigation did not reveal any non-metallic materials suitable for the temperature conditions therefore non-metallic gasket materials have not been considered further.

The ideal gasket for the application has the following general character:

1. Large cross section of simple shape for ease of fabrication and handling.
2. Reasonable in cost, as to both material and fabrication, since it is felt that, gaskets should be replaced each time the closure is opened. No re-usable gaskets have been encountered.
3. Modulus of elasticity lower than the confining surfaces.
4. High coefficient of expansion tending to produce increasing sealing forces with temperature rise.
5. Must be softer than the flange material in order that gasket surfaces are not damaged necessitating refinishing (or replaceable surfaces must be incorporated in the design).
6. Depending on flange vessel and closure design, it is desirable for the gasket to have recovery or "spring back" in order that sealing may be accomplished for all conditions of flange distortion. Since "spring back" is difficult to obtain and predict, one of the basic objects of the whole design must be to minimize or eliminate flange rotation, movement and distortion.
7. Preferably the gasket would compress between simple, easy to machine, surfaces.

Gaskets may be characterized as a solid metallic ring of various relatively simple smooth shapes. Tabulation of these along with the seating stress and required surface finish is shown on Pages 46 to 48.

All of this group of gaskets depend on high compression forces to form a seal by the indenting of their surfaces. They have no "spring back" and are not materially distorted under pressure. For optimum sealing they should be plated with a soft material like silver.

THE FLAT TYPE.

This design is relatively low in cost. Proper design will allow it to be of a large enough cross section that it will not be fragile and difficult to fabricate and install.

"O" RING TYPE (METALLIC)

This type of gasket construction can function as a ring joint gasket or on the self energized principle. In either function, this gasket will be a handling problem due to small cross-section area. There is no actual reactor test data available on this construction, however, R. L. Gastineau of United Aircraft Products, Ring Division, Dayton, Ohio, recommends a 1/2" Dia type 304 stainless steel "O" ring plated with a .002 to .003 Silver coating. Resiliency of this gasket approximately .008 to .012 Mr. Gastineau claims. Purchase of this type of gasket is not difficult and cost should be reasonable.

See Page 53 Fig. 2-3 for application.

As can be seen from the application, manufacturing tolerances are critical.

TRIANGULAR GASKET (METALLIC)

This type of construction is similar to that of the "O" Ring type, however it lends itself to the self energized principle better than an "O" ring type. The initial sealing stresses are low (200 PSI) which do not require highly stressed retaining methods. This construction being of a larger cross-section than the "O" Ring type will have some advantage over the "O" Ring type in remote control handling. Manufacturing tolerances are also critical.

GASKETS SOLID METALLIC TYPE

46

TYPE	MATERIAL	MIN. SEATING STRESS P.S.I.	SURFACE FINISH
			Concentric Serrated Ax. 800 Micro in.
Plain Solid Flat	Aluminum	8800	"
	Copper	13000	"
	Iron & St'l	18000	"
	Monel	21800	"
	Stainless St'l	26000	"

			Very Smooth 80 to 32 Micro in.
"O" Ring Solid Circular	Aluminum	8800	"
	Copper	13000	"
	Iron & St'l	18000	"
	Monel	21800	"
	Stainless St'l	26000	"

			Very Smooth 80 to 32 Micro in.
Triangular Solid	Aluminum	2000	"
	Copper	2500	"
	Iron & St'l	3500	"
	Monel	4500	"
	Stainless St'l	6000	"

SKETCH NO. 33

The following are a group of solid metallic gaskets that employ hydrostatic pressure to help achieve sealing pressure.

DELTA TYPE

Triangular in cross section, the function of this gasket is based on the unsupported area principle. The gasket being slightly wider than the combined depth of the flange grooves allows the initial sealing on the edges when retaining devices are secured. The gasket is then subjected to the internal vessel pressure to accomplish the final operating seal. In this design again manufacturing tolerances are critical and small cross section area will not allow good remote control handling. Mr. Geipel of Metallo Gasket Co., New Brunswick, N. J. advised they do not have capacity to produce 100" dia. and Mr. Todd, Melrath Co., Philadelphia, Pa. advised they do have such capacity.

RING AND LENS TYPES

Again manufacturing tolerances are critical and the cross section area to dia. ratio will not allow remote control handling. Internal pressures will tend to increase sealability. This construction also uses the unsupported area principle in which the edges form the initial seal and internal pressure forms the final seal. Manufacture of this gasket is difficult and The Melrath Co., Philadelphia is the only one found with capacity to machine 100" dia.

TYPE

MATERIAL

MIN. SEATING
STRESS P.S.I.

SURFACE FINISH 48

Ring Joint

Solid

Aluminum

2000

Copper

2300

Iron & St'l

3500

Monel

4000

Stainless St'l

6000

Very Smooth

80 to 32 Micro in.

"

"

"

"

Delta

Solid

Aluminum

1000

Copper

2000

Iron & St'l

3000

Monel

4000

Stainless St'l

5000

Very Smooth

80 to 32 Micro in.

"

"

"

"

Lens

of

Flare

Aluminum

2000

Copper

2300

Iron & St'l

3500

Monel

4000

Stainless St'l

6000

Very Smooth

80 to 32 Micro in.

"

"

"

"

SKETCH NO. 34

Solid metallic gaskets may also be of another type, those in which sealing is achieved by actually causing the gasket metal to "flow" and materially change the original shape of its cross section.

PROFILE TYPE (Page 50)

This design is similar to the solid flat gasket except that only about 50% as much seating pressure is required and better surface finishes are required. This type is being used on small reactor vessels. (Information classified). Although available in some forms the size of convolutions and materials should be engineered for the particular application. Handling is difficult, not so much because the gasket is fragile but the points of the convolutions must be preserved if reliable sealing is to be achieved. The special profile design also shown does not suffer as badly in this respect. Both types have the advantage of being able to achieve a seal regardless of thickness since sealing depends on the flow of metal from the "peaks" to the "valleys" and the forming of perfectly mating surfaces in the process. It was found that these types of gaskets are usually plated to achieve lubrication and prevent sticking.

TYPE	MATERIAL	MIN. SEATING STRESS P.S.I.	SURFACE FINISH
------	----------	-------------------------------	----------------

Solid

Stainless St'l

10,000 AX.

Concentric Serrated
Ax. 800 Micro in.

Profile

Solid

Aluminum

5500

Very Smooth
80 to 32 Micro in.

Copper

6500

"

Iron & St'l

7600

"

Monel

9000

"

Stainless St'l

10100

"

Special
Profile

Solid

Aluminum

5500

Commercial
Smooth
150 - 200 Micro in.

Copper

6500

"

Iron & Steel

7600

"

Monel

9000

"

Stainless St'l

10 100

"

SKETCH NO. 36

Other types of "flow" gaskets are shown on pages 52 and 55. Those on page 52 combine the "flow" and the exposed area principles but generally require high accuracy of manufacture and are exceptionally fragile. Further, they are designed for use with much higher pressures than 660 PSI. Very little benefit can be obtained from this principle for our application.

On page 53 are shown other types of "flow" gaskets to which the previous comments generally apply, however at the top of the page is a triangular design which permits hydrostatic forces on the closure to help with the sealing, only about 20% of final sealing forces need be initially applied by the closure retention mechanism. It does have disadvantages, there is a possibility sticking to the vessel wall and clearances must be kept quite small making assembly of the closure to the vessel difficult.

SKETCH NO. 37

SKETCH NO. 38

The configurations on page 55 are of miscellaneous gasket types of the solid metal family, one being a design which uses a very soft metal such as 50 Aluminum which is made to "flow" to such an extent that it completely fills every part of the confining cavity. This method results in relieving the closure retainer of gasket pressure. When the gasket is initially deformed it will, being a soft material, have very little "spring back" and will be extremely sensitive to any flange distortion.

Also on page 55 is shown, a novel, but untried bi-metal gasket.

As far as can be determined, this principle is new. The General Plate Division of Metal & Controls Corp., Attleboro, Mass. Messrs. Miller, Almquist & Howard submitted the following: A bi-metal ring 100" I.D. x 2" face x 3/4 thickness will deflect .060" at the center of the 2" face for a temperature change of 400°F. If this ring is confined, the expansion force will build up to more than 2,000,000#. The deflection of .060" will compensate for any distortions. The materials suggested are Stainless Steel type 304 and Invar. There is no evidence of this method ever being used and it has the disadvantage of relaxing the sealing pressure when cooled, also successive heating and cooling would tend to produce relative movement between the sealing surfaces.

SKETCH NO. 39

HOLLOW TYPES (USUALLY STAINLESS STEEL TUBES)

Very little information and no valid test data (page 57) could be found for these types. There are three variations, the hollow "O" ring containing air at atmospheric pressure, the pressurized "O" ring (a British Patent) inflated with helium to 600 PSI and the open type, where the pressure being sealed is admitted by holes or slots to help achieve a seal by expanding the "O" ring. All these types are very difficult to manufacture, (there are no vendors presently making them) require extremely close tolerances and fine surface finishes and are fragile to handle. Our information indicates that they achieve very little "spring back" and require excessively high (no quantitative figures available) seating pressures. One installation at Detroit Edison failed.

HOLLOW GASKETS

TYPE	MATERIAL	MIN. SEATING STRESS P.S.I.	SURFACE FINISH
Hollow "O" Ring Gas Filled	Aluminum	No Data	Very Smooth 80-32 Micro in.
Hollow "O" Ring	Aluminum Copper Iron & St'l Monel Stainless St'l	No Data	Very Smooth 80-32 Micro in.

SKETCH NO. 40

Another family of gaskets are the sheet metal types. They are numerous and varied, and are generally used as an attempt to gain "spring back" and resiliency. No evidence was found that, under the conditions of this problem, more than .006 to 1008 can be expected. Further, the Argonne Laboratory EBWR closure using a spirally wound "Flexitallic" gasket at best achieved only .012 "spring back". The following is a review of types available along with some comments and vendors' recommendations. As a group these gaskets are fragile and difficult to handle, especially with remote controls. See pages 60 through 62.

CORRUGATED METAL GASKET

This gasket is enclosed by a corrugated metal shell and top washer and has long been used extensively by chemical, oil and other industries where high pressures are encountered. Surface finish must be 80 micro-inch or better and bolting stresses are high. There is no reactor application data on this type, but high temperatures have an adverse affect.

CORRUGATED METALLIC ASBESTOS

There is no test data of a gasket of this construction being used for this kind of application, however, its primary use is for pressures below 600 PSI and high temperatures. Gasket requires low bolt pressures and is suitable for rough or warped flange surfaces and can be supplied in any metallic material. It is extremely fragile and is primarily intended for permanent installation.

JACKETED METAL WITH FILLER

This gasket consists of a soft filler enclosed in metallic shell covering only one face and can easily be compressed but cannot withstand high pressures. There is no test data on this kind of gasket in the reactor field. This gasket is also made with a washer covering the exposed area which will withstand temperatures to 850°F.

CORRUGATED METAL JACKET WITH CORRUGATED METAL LINER

Johns-Manville Co. suggested this type of gasket of 304 stainless steel with a raised face flange, seating stresses are approximately 7500 PSI, a fine finish (32 micro-inch) and bolting to be stressed to 20,000 PSI were also recommended. Metallo Gasket Co., New Brunswick, N. J. suggests same design. This gasket will stand compressive stresses of 15,000 PSI and over without serious damage but after high compression "spring back" is reduced to practically zero.

SPIRALLY WOUND (FLEXITALLIC TYPE)

This gasket with a "teflon" filler has been used on the Argonne EBWR. Excessive leakage was experienced. Even inner and outer diameter back up strips were not effective and over a period of continuous operation the "teflon" filler crumbled and became ineffective. Other reasons for failure were edge buckling and flange surface galling. There is no equipment in existence to manufacture this type of gasket in sizes greater than 48" diameter. (See ASTM Report 102 A)

CORRUGATED (METAL)

This gasket has merit if the following is complied with:

1. Smooth surface finish (63 micro-inch or better)
2. Compression limiting device so gasket cannot be stressed beyond its elastic limit.
3. Uniform seating (yield stress 7600 PSI) fences

The following recommendations have been made by the Melrath Supply and Gasket Co. Eight corrugations per inch with a 2" wide face and .018 thick type 304 stainless steel material with a monel strip .0312 thick spot welded to the outside diameter of gasket to act as a compression limit gauge so gasket cannot be crushed. The recommended finish of a flat flange is 63 micro-inch. The outer limit gauge strip must employ lifting ears for gasket removal.

METAL JACKETED, ASBESTOS FILLED

This type of gasket requires a smooth (at least 63 micro-inch) finish and has very little "spring back". It has been tested at the Argonne National Labs on the EBWR closure and found to be unsatisfactory.

The slightest relative movement of flanges will break the seal. It is, however, manufactured in 100 inch diameter sizes by several companies. Shipping and handling are difficult because of the small cross sectional area.

SHEET METAL GASKETS METALLIC

TYPE	MATERIAL	MIN. SEATING STRESS P.S.I.	SURFACE FINISH
Very Smooth 80-32 Micro In.
	Aluminum	5500	
	Metal Rims. Copper	6500	"
	Asbestos or Iron & St'l	7600	"
	Teflon Filler. Monel	8000	"
	Stainless St'l	9000	"
Very Smooth 80-32 Micro in.
	Aluminum	5500	
	Triumph Single Jacket Copper	6500	"
	Asbestos or Iron & St'l	7600	"
	Teflon Filler. Monel	8000	"
	Stainless St'l	9000	"
Very Smooth 80-32 Micro in.
	Aluminum	2900	
	Corrugated Jacket & Copper	3700	"
	Top Washer. Iron & St'l	4500	"
	Monel	5500	"
	Stainless St'l	6500	"

TYPE	MATERIAL	MIN. SEATING STRESS P.S.I.	SURFACE FINISH
Corrugated	Aluminum	1000
	Metal Core	Copper	1500
		Iron & St'l	2000
		Monel	2500
		Stainless St'l	2500
Spirally		Very Smooth 80-32 Micro in.
	Wound with		
	Teflon or	Iron & St'l	
	Asbestos		
	Filler & I.D. & O.D. Support Rims	Stainless St'l	
		2900	Commercial Smooth 150-200 Micro in.
		4500	

SKETCH NO. 42

SHEET METAL GASKETS METALLIC

TYPE	MATERIAL	MIN. SEATING STRESS P.S.I.	SURFACE FINISH
Corrugated Metal	Aluminum	3700	Very Smooth 80-32 Micro in.
	Copper	4500	"
	Iron & St'l	5500	"
	Monel	6500	"
	Stainless St'l	7600	"
Corrugated Metal Asbestos or Teflon insert	Aluminum	2900	Commercial Smooth 150-200 Micro in.
	Copper	3700	"
	Iron & St'l	4500	"
	Monel	5500	"
	Stainless St'l	6500	"
Deeply Corru- gated Metal Core. Partially En- closed by Metal Jacket.	Aluminum	3700	Very Smooth 80-32 Micro in.
	Copper	4500	"
	Iron & St'l	5500	"
	Monel	6500	"
	Stainless St'l	7600	"

MATERIALS

In tabular form the following are compilations of service temperatures and corrosion resistance characteristics of gasket materials for which data is available.

TABLE

Service Temperatures of Gasket Material

	<u>Material (Metallic)</u>	<u>Type</u>	<u>Max. Temperature of Continuous Service °F</u>
1	Tin	--	212
2	Lead	--	212
3	Zinc	--	212
4	Magnesium	--	400
5	Admiralty Brass	--	500
6	High Brass	--	500
7	Copper	--	600
8	Evordor	--	600
9	Aluminum	--	800
10	Stainless Steel	304	800
11	" "	316	800
12	" "	347	1700
13	Reme Iron	--	1000
14	Armco Iron	--	1000
15	Low Carbon Steel	--	1000
16	Silver	--	1200
17	Gold	--	1200
18	Chrome Moly. Steel	502	1200
19	Chrome Steel	410	1300
20	Nickel	--	1400
21	Monel	--	1500
22	Inconel	--	2000
23	Hostelloy	"A" or "B"	2000
24	Platinum	--	2300
25	Tantalum	--	3000
<u>Material (Non Metallic)</u>			
	Asbestos (white)	Chrysotile	900
	" (composition sh't)	Rubber Base	850
	" (blue)	Crocidolite	900-850
	Rubber	Natural Base	212
	Syn. Rubber & Plastics		250
	*Teflon (Polymerized tetrafluoroethylene)		617

Ref Johns-Manville Vol. 1 No. 5 - 6 Chemical & Heat Resistance of Gasket Materials.

*Not recommended for gasket application above 392°F or for high pressures unless confined in a tongue & groove or similar joint.

TABLE
Corrosion Resistance
of
Gasket Materials

Resistance Rating A - Good F - Fair C - Depends on Condition X - Unsuitable	Metals									Asbestos			Teflon	
	Lead	Copper	Aluminum	Monel	Nickel	Iron & St'l	St'n. St'l. 304	St'n. St'l. 316	St'n. St'l. 347	White (comp. or woven)	Blue (comp. or woven)	Compressed Sheet		
SOURCE OF DATA	G	G	G	G	G	G	G	G	G	J	J	U	E	P
Steam <500°F	C	A	A	A	A	A	A	A	A	A	-	A	A	-
500°F-1000°F	X	C	C	C	C	A	A	A	A	A	-	C	X	-
>1000°F	-	X	X	X	X	X	A	A	A	C	-	X	X	-
Water (fresh	A	A	A	A	-	C	A	A	-	A	-	A	A	A
(return condensate)	A	A	A	A	-	A	A	A	-	A	-	A	A	A

Sources of data:

- (G) Goetze Gasket & Packing Co.
- (E) E.I. du Pont de Nemours & Co.
- (J) Johns-Manville Corp.
- (P) The Pfaunder Co.
- (U) United States Rubber Co.

Gasket Surface Finish requirements are an important consideration. The best information reveals little data except the following definitions and test results.

DEFINITIONS:

1. "Very Smooth" .80 micro-inch RMS or better.
2. "Commercial Smooth" .0005 deep groove with 80 to 100 serrations per inch of width. 150-200 micro-inch RMS.
3. "Commercial Concentric Serrated". Concentric grooves approximately $1/64$ " deep and no more than $1/32$ " apart. Tips of ridges should be uniform in height & sharpness. If tip exceeds a width of .010" effectiveness of face reduced.
4. "Commercial Phonographic". Continuous spiral cut approximately .010 deep at rate of 32 serrations per inch of width. Tips of ridges should be uniform in height & sharpness. Tip width should not exceed .010. The contour of groove should be "U" rather than a sharp "V".

It will be noted from charts on page 66 that flat gaskets seal best on rather rough surfaces and corrugated and spiral wound metal gaskets seal best on smooth surfaces.

FIG. 1.

FIG. 2.

FIG. 3.

SKETCH NO. 44

COMMENTS:

The curves in Fig. 1 Page 66 indicate that the best surface finish for a flat metallic gasket is the concentric grooved type (G) with sharp, spaced ridges. The wedging action forces the gasket material to flow sufficiently to seal.

The curves in Fig. 2 Page 66 indicate that the best results, for a corrugated steel jacketed asbestos filled gasket, was the smoothest surface finish. As can be seen even the shallow fine groove of facing B reduced sealability considerably.

The curves in Fig. 3 Page 66 indicate that the surface finish has little effect upon sealability of the spirally wound gasket. However, curve A indicates good sealing at lowest gasket stress.

LEAKAGE DRAIN OFF

The amount of leakage (5 lbs. per day) is small and it is suggested that this be accomplished by draining by gravity to a tank either vented closed depending on the method used to isolate the closure from the pool water. If the air pressure method is used, the leakage drain tank will be exposed to the same pressure and could not be vented to atmosphere.

In the region of the gasket or gaskets leak off must be drained before it reaches the outside of the vessel. Probably the best way to accomplish this is to incorporate two gaskets (The secondary one need not be more than a static water seal) with the leak off between. As a safety measure two identical pressure gaskets may be used or a wide single gasket with communicating drains in its central section.

See page 69.

SKETCH NO. 45

REGISTRATION

The problem of registering the closure with the parts inside the vessel breaks down into two parts; radial and central.

Both central and radial location may be achieved by the use of two studs as dowels in a flange and stud arrangement, by the use of pins and holes in a vertical plane on various members and by keys and grooves or pins and sockets in projections for the purpose either on the outer periphery of the flanges or attached to the inside surfaces of the vessel and the closure.

Central location of the closure can be arranged allowing the wall of the closure to project into the vessel (as in the self energized gasket design) and making it thin enough that hydrastatic pressure will expand it to contact the vessel walls, centralizing the closure. This allows the closure wall to be some .250 smaller in diameter than the inside diameter of the vessel. It is to be noted that as the larger this clearance can be made, the less danger of "freezing" and greater the ease of assembly.

POOL WATER EXCLUSION

It is necessary to exclude the pool water from direct contact with the closure to prevent a high temperature differential across the closure wall. There appears to be two methods of accomplishing this. One, lagging the entire enclosure and encasing with a stainless steel, water tight shield. This approach becomes complicated and cumbersome since some sorts of openings and water seals are necessary in order to assemble and disassemble the closure.

The second possibility is a large inverted bell of light stainless clad steel lowered over the whole closure and held down to the pool floor, Sketch 72. Air pressure is then introduced to expell the water from the bell and acts as the insulating medium. Size of the bell can be reduced by incorporating insulation in the walls of the bell and introducing a compression seal around the cables. Cables can be brought out thru the top as sketch 73 or brought out through a channel under the edge of the bell. It will be noted that with proper piping the same pipe may be used for introducing and bleeding the pressurizing air.

REMOTE HANDLING

Although methods of remote handling have been considered, no attempt is made to suggest design and they are mentioned only in the following list and in connection with their possibilities of application to some closure arrangements.

- Stud stretching devices.
- Long-handled socket wrenches with or without built in torque measuring devices.
- Hooks and levers with long handles for operating latches, moving and placing parts, pneumatic or hydraulic cylinders, pumps and valves.
- Magnetic and electromagnetic devices used in place of hooks and clamps.
- Submersible electric motors.

A77

SKETCH NO. 46

A78

SKETCH NO. 47

73

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Design Specification
for the
CRBR Pressure Vessel Closure

Scope

This specification covers design and performance requirements for a quick opening closure. The closure will be part of the pressure vessel assembly which houses the reactor core and other components.

References and Codes

1. 43-301-195	Reactor Section
2. 43-001-875	Proposed Reactor Vessel and Pump Layout
3. A.S.M.E.	Boiler and Pressure Code for Unfired Pressure Vessels, Section VIII, and Section IX, 1956 Edition.

General Description

The nuclear heat source within the vessel is cooled by the flow of boiling water passing directly through uranium bearing fuel elements. Drawing 43-301-195 shows the arrangement within the vessel. The external arrangement is shown by drawing 43-001-875. The reactor vessel will be 11'-0" inside diameter with an overall metal wall thickness of approximately 3". A pool of pure deionized water above the reactor approximately 18 feet deep, serves as a radiation shield during reactor operation and refueling periods.

Closure Requirements

The removable reactor head must be removed every four months to perform refueling operations. The closure must provide a minimum clearance diameter of 7'-3".

The upper surface of the head shall be suitably protected by sealed thermal insulation to prevent contact between the cool pool water and the hot surface of the vessel. The insulation shall be considered an integral portion of the removable head. The closure design shall prevent any contact between the pool water and the vessel.

The main closure seal must be designed to contain the full design pressure of the vessel (660 psig at 496°F). The leakage through this seal shall not exceed 5 lbs/day. A secondary seal must be provided to prevent the entrance of water from the shield pool above the vessel. Erosion or cutting of the vessel gasket surfaces, due to leakage, shall not be tolerable.

A segmented ring, breech lock or other type of device shall be used to lock the cover in place during operation. The target time to assemble and disassemble the locking device shall be 30 minutes.

Heavy metal sections in the closure should be avoided so that the vessel may be brought up to temperature (100°F to 496°F) or cooled off (496°F to 100°F) in approximately one hour without introducing excessive thermal stresses.

The closure design must be such that it conforms with the Unfired Pressure Vessel Code, or close enough to the code requirements that special ruling can be obtained.

Twenty (20) control rod drives will be mounted on the vessel cover. The cover shall contain a suitable indexing device to position the control rod drives with respect to control rod guide tubes located within the vessel.

It is, of course, understood that all manipulations required to open the closure must be performed under 18 feet of water with long handled tools. Water free from rust for visibility.

The closure design must be inherently free of any tendency to "freeze".

Design Data

1. Design pressure	660 psig
2. Design temperature	496°F
3. Normal operating pressure	600 psig
4. Hydrostatic test pressure	990 psig
5. Vessel wall thickness	3" total
a) Base metal SA 212-B	2-3/4"
b) Stainless clad 304 ELC	1/4"
6. Vessel steel I.D.	11'-0"
7. Minimum vessel closure opening	7'-3"
8. Max. heating & cooling rates	400° F/Hour
9. Weights of various components	
a) Control rod drive mechanisms	8,500 lbs.
b) Vessel cover	Approx. 16,000 lbs.
10. Normal shield pool temperature	100°F
11. Depth of shield pool water above closure	18 ft.
12. Shock load on vessel cover during control rod scram	80,000 lbs.
13. Maximum permissible misalignment between control rods and guide tubes.	± 1/32"
14. Internal heat generation due to radiation at the inside surface of the cover	400 BTU HR-FT ³

JP:gw

Jack Patterson
Nuclear Power Division
7/24/1958

SOME GASKET SUPPLIERS:

METALLO GASKET CO.
10-18 Bethany Street
New Brunswick, N. J.

AKRON METALLIC GASKET CO.
Akron, Ohio

MELRATH SUPPLY & GASKET CO.
Tioga & Memphis Street
Philadelphia 34, Pa.

FLEXITALLIC GASKET CO.
8th & Baily Street
P.O. Box 680
Camden, N. J.

JOHNS-MANVILLE CO.
(GOETZE GASKET DIV.)
22 East 40th Street
New York 16, N.Y.

Appendix B

QUICK OPENING CLOSURE FOR CRBR REACTOR VESSEL

This report covers work performed under Allis-Chalmers Manufacturing Company's Purchase Order No. WA-440304-NSP, a part of the Research and Development Program under AEC Contract No. AT(11-1)-589 with Northern States Power Company.

Koerper Engineering Associates, Inc.
7414 West State Street
Milwaukee 13, Wisconsin

Mr. Clifford Graham, Chief Engineer
Nuclear Power Division
Allis-Chalmers Company
West Allis, Wisconsin

August 28, 1959

Subject: Quick Opening Closure for CRBR Reactor Vessel
KEA Report No.-KDM-AC-1011 P.O. No. WA-440304-NSP

Dear Mr. Graham:

Herewith is our completion report covering our study of mechanisms and seals for the rapid opening and closing of the reactor vessel. The report covers methods and analyses submitted by KEA and also comments on various systems proposed by Allis-Chalmers.

Briefly the conclusions arrived at by KEA and substantiated herein are -

1. The most rapid and possibly the most economical closure construction includes a self-energizing type of seal.
2. Reinforced Teflon of the materials now available offers good possibilities for the specified service.
3. To assure a satisfactory seal design, a development program on seal materials and construction is recommended herein.
4. In the absence of conclusive test results on self-energizing seals, a conventional prepressurized gasket seal is indicated.

A very effective working relationship has been developed with your staff members. In our contacts with them they have been most cooperative.

This report completes KEA's assignment on the closure analyses.

Cordially,

E. C. Koerper
President

Att: KDM-AC-1011

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Introduction and Summary

The purpose of this investigation has been to thoroly study closure designs as prepared by A-C and proposed by KEA in order to arrive at a suitable combination for use in the reactor vessel. This comprehensive analytical approach has been made to assure early elimination of less satisfactory combinations and to concentrate on practical and economical solutions.

The following factors have been evaluated in each of the designs for an optimum solution.

1. Positive sealing - imperative
2. Trouble-free mechanism
3. High speed operation
4. Long life
5. Economy of fabrication.

The highest preference has been given to a complete sealing which is imperative and includes for added safety 2 seals in series with a weep channel between them.

While most analysed designs seem to be more or less workable from a mechanical point of view, the seal problem remained difficult. Locking wedges, cams or sectors, tho generally positive in action and position, do not guarantee a positive seal without satisfactory seal pressure. An answer to positive sealing with high speed operation will be a resilient self-energizing seal. This is not readily available at this time.

Introduction and Summary - (Con't.)

Further investigation and analysis of self-energizing high pressure seals is recommended. Its results can lead to a faster, safer and cheaper construction of a reactor vessel closure. Teflon offers one of the best of the material possibilities at this time. Simple test equipment is specified in this report.

In the absence of conclusive test results on self-energizing seal materials and mechanisms, a more conventional prepressurized gasket seal is indicated.

Project Assignment

The new concept of economical power generation from nuclear fuel as expressed in the Allis Chalmers Controlled Recirculating Boiler Reactor is in part dependent on minimum shut-down time for refueling. This in turn necessitates a rapidly operable closure which will seal positively against steam at 600 psi and 496° F to hold down a "lid" approximately 7 ft. 6 in. in diameter against a total pressure of about 4-1/4 million pounds. The target time for locking or unlocking the closure is 30 minutes.

This assignment is to provide the analysis, conceptual design, layout and general specifications in order that closure prototypes can be built and tested in preparation for use in the final reactor vessel. The scope of the assignment to KEA is covered by Allis Chalmers, Nuclear Power Division's memo dated September 19, 1958.

Our analytical approach has been comprehensive to assure preliminary consideration and evaluation of all pertinent factors. In turn this permits early elimination of the less satisfactory combinations and focus on the best possibilities.

Tho this reactor is in part developmental, we have placed considerable emphasis on simple, practical and economical solutions. These features are variously incorporated in the designs presented herewith.

In order for the final closure design to be most successful, it is important that the following factors be given careful consideration for an optimum solution.

1. Positive sealing.
2. High speed operation.
3. Trouble-free mechanism.
4. Long life.
5. Economy of fabrication.

To be more specific, the following points are considered:

1. Two high pressure seals in series might be desirable for positive sealing. A weep channel between the seals should be provided.
2. Minimum time of both opening and closing is important.
3. Conceptual approach should have a recommended design and alternatives having in mind possible modification convenience.
4. Gasket construction should preferably be self-energizing.
5. Gasket seal construction should be economical and of practical materials and construction.
6. Self-alignment design of closure is necessary for meeting center line specifications of control rods.
7. Closure components should be economically machined, preferably mostly on cover instead of on vessel.
8. Operating control should be positive and operate with minimum number of openings thru the insulating shell.
9. Low pressure secondary sealing is to be provided between the thermal insulation and the vessel to exclude shielding water.

10. A passage is to be provided between the high pressure seals and the low pressure seals to permit draining of water andor elimination of steam generated in the enclosure between the two types of seals.
11. Metal masses of closure cover and ring should be minimum for expedient cooling.
12. Closure cover should be of minimum weight for ease in handling.

Inasmuch as all operations must be accomplished rapidly and remotely under 18 ft. of protective water and thru a thermal insulating shield, the following factors must be considered for a desirable mechanism:

1. Overall target time is 30 minutes for opening and closing the reactor cover.
2. Mechanisms should be simple for trouble-free operation.
3. Operation to be jam-proof and/or to have alternate method of opening.
4. Minimum operating force requirements are desirable to reduce distortion of components.
5. Moving components to be preferably small, for small forces and ease in handling.
6. Controls should be readily accessible.
7. Minimum sliding contact pressures to prevent gauling and keep down operating forces.
8. There should be no loose parts to become detached and fall off.
9. Minimum requirements for special tools and auxiliary equipment.
10. Indication of position of locking mechanism is desirable.

Materials to be used have special significance in this application. They fall within 2 categories:

- 1 - Metallic for structural use, and
- 2 - Organic or metallic base for sealing purposes.

The following are factors for consideration:

1. The application is to be considered as water and/or water-vapor phase exposure with radiation exposure.
2. Low corrosion rates are important at adverse temperatures and pressures and ambient conditions.
3. The material in its location should be selected for low electrolytic corrosion combinations.
4. Corrosion by-products should be isolated from the nuclear system and preferably "non-poisoning".
5. Materials should have adequate resistance to radiation damage.
6. Design should reduce stress corrosion conditions.
7. The sliding materials and bearings should preferably operate without lubricants that might wash off.
8. Sliding contacts under pressure should be avoided, if possible, to eliminate gauling or possible blind gouging of softer and more machinable materials.
9. Materials should preferably be economical, be readily machinable and have long life.

As requested we have commented herewith on the following 5 closure designs submitted to KEA for analysis.

<u>Page</u>	<u>Type</u>	<u>AC Dwg. No.</u>
A-3	Cam Lock Closure	43-001-884
A-4	Lug Operated Shear Plate Closure	43-001-888 43-001-889
A-5	Air Operated Cam Closure	43-001-915
A-6	Rotating Lug Closure	43-001-916
A-7	Bolted Closure	43-001-917

Page A-2 lists the major factors considered in this analysis.

Only sufficient time was spent on this phase of the assignment to make a preliminary comparison of feasibility between the designs listed.

It is our opinion that improvement can be made over the above arrangements by incorporating into a single design more of the desirable factors listed on Page A-2, (9).

Primary Seal

A - Locking Method - Bolt, Clamp, Radial Pins, Breech Lock, Shear Ring, Segmented Ring, Bolted Flange, Air operated Cam, etc.

B - Auxiliary Equipment Required - External Remote Control

C - Locking Forces - Main - Shear, Tension or Compression

D - Speed of Operation - Time

E - Forces on High Pressure Seal

External

Internal

Pretension

Material - Resilient

Floating

Structure

Internal

Pressure Expansion

F - High Pressure Seals - Preferably 2.

Location - in lip of vessel with drain channel

Material

Construction - to require minimum space

Replacement and Auxiliary Equipment Needed

G - Simplicity of Construction

H - Original and Subsequent Machining

I - Overall Costs

Secondary Seal

J - Location

K - Material

L - Construction and Action

M - Replacement and Auxiliary Equipment

N - Thermal Insulation Considerations

O - Overall Costs

Advantageous Features

Low torque requirements to positively lock and unlock sealing cam.

Lock mechanism easily replaced.

Questionable Features

Locking mechanism is complex requiring numerous parts which can drop off into the reactor as the cover is being removed.

The high pressure seals are located in the cover making them difficult to replace.

Two secondary low pressure seals would be expensive.

The secondary high pressure seal bellows is precarious.

The many bolts on the secondary high pressure seal would need pre-tensioning and high uniform take-up torque. O-Ring seals must be provided for each bolt.

The insulation protection above the 2 secondary high pressure seals lacks effectiveness.

The low pressure seal design does not accommodate variations of machinery or thermal expansions.

Drain channels not shown.

Overall cost relatively high.

• LUG OPERATED SHEAR PLATE CLOSURE
Allis Chalmers' Dwg's: 43-001-888 and 889
Similar to 43-001-884

Report: KDM-AC-101
Date: Oct. 12, 1958
Page: A-4 - 12

Advantageous Features

Locking lug and shear plate are positive in action and position in locking.

Locking mechanism is easy to replace if lift rings are provided.

Questionable Features

High pressure seals are in lid or cover and difficult to get to for replacement.

Primary high pressure seal requires an expander for operation which is not shown.

The secondary high pressure seal is complex and too far away from the primary high pressure seal.

Secondary high pressure seal requires pretensioning by numerous bolts which is slow in operation.

Drain channels are not provided.

Each of the pretensioning bolts requires O-Ring sealing.

Secondary seal is simple and requires no mechanical fastening, but will not accommodate to dimensional change of the cover.

Overall cost is relatively high. Comments on 43-001-884 generally apply here.

Advantageous Features

Locking operation fast. Pneumatic operating system simple.

Minimum forces required to operate cam.

Total pretensioning pressure is small inasmuch as the pressure is needed only to overcome the unit pressure on a narrow annular ring.

Questionable Features

Cylindrical surfaces of the cam are difficult and expensive to machine. The tolerances of the numerous pocket seats are critical.

Pneumatic system is not positive requiring second set of cylinders..

The primary high pressure seal is small and solid and located in the cover and would have to be cemented in place under water.

The annular seal ring must flex under working pressure to be effective.

The primary seal lip need not be both screwed down and welded to the vessel. Only welding would be sufficient.

High pressure seals are in the cover.

O-Ring on bottom of pretensioning screw needs to be retained.

Each screw and nut requires another O-Ring for sealing.

There is no positive indication of cam position.

Weep channels for water and steam are desirable .

Low Pressure seal O-Rings have limited tolerance for sealing.

Advantageous Features

Rotating cam lock is rapidly operable with few tools.

Other Features

Tho the locking and the unlocking of the cam is relatively fast,
the pretensioning of the ring is relatively slow.

Insufficient percentage of area of the lip is effective against
the rotating lug.

Comments on high pressure seals are the same as those expressed for
cam lock drawing 43-001-915.

The rotating cam can not operate as drawn.

The projected cam area for both shear and compression is small and
inadequate.

The cam lock shaft is not sealed against shielding water.

The pretensioning screw shaft gland need not be so complex.

The overall cost of the closure mechanism is relatively expensive.

Advantageous Features

Bolted closure is relatively simple in construction and inexpensive.

Questionable Features

Approximately 75 bolts 3 inches in diameter are required. This requires considerable pretensioning making the operation rather slow and requiring considerable auxiliary heavy equipment.

The primary seals are in compression only and would tend to leak sooner than self-energizing seals.

Seals are not self-energizing and are difficult to locate on an unrecessed surface.

Bolts projecting into the shielding water require sealing and also insulation to conserve heat flow to shielding water.

Welding of positioning stud to vessel is questionable. To do otherwise necessitates heavy sections.

Secondary low pressure seal is inadequate for the purpose and will not accommodate to thermal changes.

The drain and bleed system is complex.

THERMAL-RING CLAMP CLOSURE
Preliminary Analysis of A-C Design

Report No. KDM-AC-1011
Date: Nov. 28, 1958
Page: 1 - 21

For purposes of applying a preload to the gasket surface and to compensate for any tolerances, the ring would be expanded by heating with cal-rods and then allowed to cool and contract while being engaged with the cover and reactor vessel flange.

The ring material would be low carbon steel and it would weigh approximately 3,000 lbs. A specific heat of 0.12 and a linear thermal elongation coefficient of .00000636 in. per in-F were used for the material.

The section to be elongated was taken as 9 in. long and 1.25 in. thick on a 96 in. inside diameter. The elongation by thermal expansion was assumed to be .040 in.

The temperature of the surrounding medium and the vessel was taken as 100° F.

A temperature rise (ΔT) was calculated as 700° F would be required for an elongation of .040 in. in a 9 in. length. The heat required for this temperature rise of 700° F in the steel ring was determined as 252,000 BTU.

The losses which would occur during the heating cycle of approximately 20 minutes were calculated as 2.898×10^6 BTU.

The total quantity of heat required would be 3.15×10^6 BTU. In terms of electrical power this amounts to 925 KW hrs. There are limitations to the amount of power which could be applied. Assuming that all of the available area could be utilized on the outer surface of the ring, this surface would be covered with 1500° F cal-rods. The quantity of power that could be applied equals 216 KW hrs.

THERMAL-RING CLAMP CLOSURE

• Preliminary Analysis of A-C Design

Report No. KDM-AC-1011

Date: Nov. 28, 1958

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The amount of power required is high and is approximately four times the amount that can be applied. Therefore, it is apparent that this method of clamping does not appear to be feasible.

Although the thermal ring appears impractical at the present time based on rough calculations, it may have possibilities if used in conjunction with some mechanical means by which the amount of thermal elongation could be minimized. This would reduce the temperature differential required and thereby reduce the losses incurred.

Other variables as the length of section, the material and insulators could be investigated and advantageously revised to contribute to the feasibility of the clamp.

One main disadvantage with this design is the lack of an alternate method of removal in case of failure to obtain the proper heat.

J. Melter
Project Engineer

• REACTOR VESSEL CLOSURE ANALYSIS
Thread Closure Calculations

Report: KDM-AC-1011
Date: Oct. 21, 1958
Page: 3 - 23

Coefficient of friction factor given for high compressive loading.

Coefficient of sliding friction .15(u)

Coefficient of static friction .25(u)

P = Actuating force (lb.) @ the pitch radius

L = Load applied by the screw

m = Pitch (or lead) of the thread = 1 inch

r_1 = Pitch radius of thread = 47.75 inches

r_2 = Mean friction radius = 47.00 inches

P_o = Force required with no friction = $L(\frac{1}{300}) = L(.0033)$ Lb.

e = Efficiency

Total Load "L" Applied by Screw Equals $W_1 + W_2$ where:

W_1 = Pretension equal to internal load caused by 660 psi(p_1) steam pressure on 90 inch diameter (inside) head.

W_2 = Gasket load of 10,000 psi (p_2)

$$W_1 = p_1(\frac{\pi D^2}{4}) = 660(\frac{\pi(90)^2}{4}) = 4.2 \times 10^6 \text{ lbs.}$$

$$W_2 = p_2(c)(w) = 10,000(290)(L) = 2.9 \times 10^6 \text{ lbs.}$$

$$L = W_1 + W_2 = 4.2 \times 10^6 + 2.9 \times 10^6 = 7.1 \times 10^6 \text{ lbs.}$$

For Motion in Same Direction as "L" Load

$$P_1 = L \left(\frac{2\pi u r_1 - m}{2\pi r_1 + um} \right) + L \frac{r_2 u}{r_1}$$

$$P_1 = L \left(\frac{300(.15) - 1}{300 + .15} + \frac{47.00}{47.75}(.15) \right)$$

$$P_1 = L(.14659 + .14764)$$

$$P_1 = L(.294) \text{ Lbs.}$$

$$P_1 = 7.1 \times 10^6 (.294) = 2.09 \times 10^6 \text{ lbs. (in direction of load "L")}$$

$$e = \frac{P_o}{P_1}$$

$$e = \frac{L(.0033)}{L(.294)} = .0113$$

$$e = 1.12\% \text{ Efficiency}$$

For Impending Motion in Opposite Direction as "L" Load

$$P_2 = L \left(\frac{2\pi u r_1 + m}{2\pi r_1 - m} \right)$$

$$P_2 = L \left(\frac{300(.25) + 1}{300 - .25} + \frac{47}{47.75}(.25) \right)$$

$$P_2 = L(.25354 + .24607)$$

$$P_2 = L(.49961) \text{ Lbs.}$$

$$P_2 = 7.1 \times 10^6 (.49961) = 3.55 \times 10^6 \text{ lbs. (Opposite Direction of Load "L")}$$

$$e = \frac{P_o}{P_2}$$

$$e = \frac{L(.0033)}{L(.49961)} = .0066$$

$$e = 00.66\% \text{ Efficiency}$$

Hydraulic Cylinder Size Requirement

Note: The maximum actuating force is 3.55×10^6 lbs. (1725 tons).

$$N = 4 \text{ Hydraulic Cylinders @ } 5,000 \text{ psi } (p_3)$$

$$A_T = \text{Total Area}$$

$$A = \text{Cylinder Area}$$

$$d = \text{Diameter}$$

$$H = \text{Length of Stroke}$$

$$H = 20^\circ/360^\circ \times 300 \text{ in.} = 16.75 \text{ in. (Approximately)}$$

$$A_T = \frac{P_2}{P_3} = \frac{3.55 \times 10^6}{5 \times 10} = 710 \text{ sq.in.}$$

$$A = \frac{A_T}{N} = \frac{710}{4}$$

$$A = 177 \text{ sq.in.}$$

$$d = \sqrt{\frac{A(4)}{\pi}}$$

$$d = 15 \text{ in. (Diameter of each cylinder)}$$

Total Volume (V) of Hydraulic Fluid Required

$$V = 710 \times 16.75$$

$$V = 11,900 \text{ cu.in.}$$

$$V = 51.5 \text{ gallons}$$

CONCLUSIONS

Four 15 inch diameter cylinders operating at 5,000 psi would be required to supply an actuating force of 1,725 tons to loosen this closure. Somewhat less force is required for closing.

The apparent mechanical advantage is 300 to 1, if neglecting friction. When the effects of friction and the low efficiency of the thread are also considered, the actual mechanical advantage is approximately 2 to 1.

The load applied by the screw on the flange is made up of two parts, (a) the pretension to counteract the steam load on the cover, and (b) the load to compress the gasket, the total being equal to 7.10×10^6 lbs.

There are high stress concentrations in the lower section of the clamping ring at the corner. These could be eliminated by adding a radius in the corner and increasing the section.

The stop on the lower flange should be effective in both directions because the frictional force between the rings lower flange and the vessel flange is not great enough to resist turning when the actuating force is applied to loosen the closure.

• REACTOR VESSEL CLOSURE ANALYSIS
Wedge Segment Closure Calculations

Report: KDM-AC-1011
Date: Oct. 21, 1958
Page: 6 - 27

Coefficient of friction factor given for high compressive loading.

Coefficient of sliding friction .15(u)

Coefficient of static friction .25(u)

P = Actuating Load

L = Load applied by the wedges

B = Wedge Angle = 8 degrees

A = Friction Angle

N = Number of wedges

e = Efficiency

m = Pitch (or Lead) of thread = .25 inch

Q = Torque

F = Force on the screw

r_1 = Pitch radius of screw = 1 inch

r_2 = Mean friction radius

Total Load "L" Applied by Wedges Equals $W_1 + W_2$ where:

W_1 = Pretension (none required)

W_2 = Gasket Load of 10,000 psi (2.9×10^6) lbs.

$L = W_1 + W_2 = 2.9 \times 10^6$ lbs.

• REACTOR VESSEL CLOSURE ANALYSIS
Wedge Segment Closure Calculations

Report: KDM-AC-1011
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Force Required to Drive Wedge

$$P_1 = \frac{L}{N} \tan (B + 2A)$$

$$P_1 = \frac{L}{N} .14054 + 2(.15)$$

$$P_1 = \frac{L}{N} .44054$$

$$P_1 = \frac{2.9 \times 10^6}{20} (.44054)$$

$$P_1 = 6.38 \times 10^4 \text{ lbs.}$$

Force Required to Loosen Wedge

$$P_2 = \frac{L}{N} \tan (2A - B)$$

$$P_2 = \frac{L}{N} (2 (.25) - .14054)$$

$$P_2 = \frac{L}{N} (.44946)$$

$$P_2 = \frac{2.9 \times 10^6}{20} (.44946)$$

$$P_2 = 6.51 \times 10^4 \text{ lbs.}$$

Wedge Efficiency

$$e = \frac{\tan B}{\tan (B + 2A)} = \frac{.14054}{.44054}$$

$$e = 31.8\% \text{ Efficiency}$$

Screw Load Required

$$F = P_2 \left(\frac{m + 2\pi r_1 u}{2\pi r_1 + u m} \right) + P_2 \left(\frac{r_2}{r_1} \right) u$$

$$F = 6.51 \times 10^4 \left(\frac{.25 + 2\pi(.25)}{2\pi(1) + .25(.25)} \right) + \text{Negligible Quantity}$$

$$F = 6.51 \times 10^4 \left(\frac{1.82}{6.35} \right)$$

$$F = 1.82 \times 10^4 \text{ lbs.} = 18,200 \text{ lbs.}$$

$$Q = F \times r_1$$

$$Q = (1.82 \times 10^4)(1)$$

$$Q = 18,200 \text{ in.lbs. (Torque)}$$

Screw Efficiency

$$F_o = P_2 \frac{m}{2\pi r_1}$$

$$F_o = 6.51 \times 10^4 \left(\frac{.25}{2\pi(1)} \right)$$

$$F_o = 2590 \text{ lbs.}$$

$$e = \frac{F_o}{F}$$

$$e = \frac{2590}{18,200}$$

$$e = 14.2\% \text{ Efficiency}$$

Overall Efficiency

Note: The efficiency of the Bevel Gear set is taken as 95%.

$$e_o = e_g \times e_w \times e_s$$

$$e_o = (.95)(.318)(.142)$$

$$e_o = 4.30\% \text{ Overall Efficiency}$$

CONCLUSIONS

The actuating force, in the form of torque, is relatively small (18,200 in.lbs.) for each segment, of which there are 20.

With the members in direct shear, no pretension is required other than the force required to compress the gasket.

This closure can be used with the compression packing or self-energizing seal.

The efficiency of this closure is 600% greater than the thread closure.

The wedge segment and drive parts are relatively small and, therefore, easy to machine.

The cover could be designed to reduce or eliminate the tendency of gasket scuffing, however, scuffing is not critical if a self-energizing seal is used.

Out of the study and proposal of various mechanisms, arrangements and materials, KEA has arrived at a number of combinations which are recommended for further consideration of their respective features.

All the arrangements include self-energizing seals.

Without some development and test it would not be profitable to attempt more precise design. For this reason comments are made on the attached layouts without stress values.

Closure Design on page 41 is submitted in a form to emphasize the features we feel are attainable with a seal material development program as suggested herein.

All of the proposed arrangements are presented in schematic form and are not necessarily in design form.

Wedge Type Closure - Screw-Gear Actuated

Advantageous Features

Locking wedge is positive in action and position. No loose parts to handle when cover is being removed. All wedges could be unlocked at the same time by means of a common gear drive which would result in a rapid completion of the opening-closing cycle. Seals remain in vessel lip during removal of cover. The miter gears offer further possibility of increasing mechanical advantage.

Questionable Features

Complex locking mechanism requires numerous parts. The tolerance of the engaging flange is critical. An even down-pressure may not be achieved due to varying friction between wedges and wall. A series tensioning member to accommodate to this variation may be necessary. The power requirement for a control operated opening-closing is high. The self-energizing seals require a minimum of pretensioning.

Schematic Only

KEA Rotating Sector Type

Advantageous Features

Locking operation is positive and fast and may be operated by a common source, tho this would complicate the structure. Relatively small forces are required to operate this device. Can be quite economical and fast in application.

Questionable Features

The tolerance of the engaging flange is critical. Preloading of the 2 pressure seals cannot be obtained by this method. Self-energizing seals are necessary to provide better sealing.

KEA Rotating Sector Type With Separate Seal Ring

Advantageous Features

Fast and positive locking operation, which may be centrally operated. Minimum forces required for locking. Requires self-energizing wedge type seals. Economical and simple in construction.

Questionable Features

Critical tolerance and possible warping of the engaging flange. Here, when there are 2 seals in parallel, the smaller diameter or inside seal will not prevent any leakage. The seal retaining ring is centered between the 2 seals, thus reducing the contact pressure where it is needed. The retaining ring will tip, reducing the possibility of initial sealing. No weep channel is possible. Difficult to put in place and may lose gaskets.

• HIGH PRESSURE REACTOR VESSEL CLOSURE
KEA Rotating Sector

Report: KDM-AC-1011
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Sectors in pairs

Schematic Only

• HIGH PRESSURE REACTOR VESSEL CLOSURE
KEA Rotating Sector With Separate Seal Ring

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KEA Eccentric Cam

Advantageous Features

This arrangement provides for a large contact area and is rapidly operable. Low force requirement in operation because of self-energizing seal. Wedge slots are provided to move a segment by pinch bar should it become stuck.

Questionable Features

There is insufficient force for preloading of the 2 seals, which, therefore, requires self-energizing seals. Overall cost is relatively expensive.

KEA Eccentric Cam Sliding at 45°

Advantageous Features

Fast and positive operating. Simple construction. 45° slide guides are mainly in direction of applied force, which reduces possibility of jamming.

Questionable Features

Preloading of pressure seals is not feasible and will, therefore, require self-energizing seals.

KEA Lever Operated Wedge

Advantageous Features

Rotating cam locks rapidly with few tools. No loose parts to handle when cover is being removed.

Questionable Features

Insufficient area between slot and pusharm. Wedge sector will have tendency to wedge thru rotation. The torque requirement for a central operation is high.

Schematic Only

Schematic Only

Schematic Only

The sketch on page 42 shows the closure design principle recommended by KEA. This includes two self-energizing seals of the type proposed earlier for testing.

The seal grooves are located in the vessel lip adjacent to a flange which accurately sets the compressed seal height. A weep channel between these seals enables checking on the amount of leakage.

Because of the feature of self-energizing, the amount of precompression is reduced and the forces required to push or pull the segments into position will be relatively low. The load applied to the segments by the cover due to internal pressure is held in shear and bending of which the forces are not excessive. The lock segments should have a slight taper to be self-locking and to insure proper contact between the components to be locked.

Final details on how to best actuate this closure are not shown. This will depend in part on the characteristics developed in the seal test program. Some of the possibilities are shown in the foregoing illustrations.

The secondary low pressure seal should be a flexible type to accommodate to thermal expansions and to machining variations in the structure.

Schematic Only

Seals for this service application fall into two main categories, namely -

1. Compression Gasket Seals - requiring high compressive load to seat the material leakproof in its groove. These materials have little resilience and must, therefore, be precompressed to retain sufficient pressure when the compression member, often a screw, extends under tension.
2. Self Energizing Seals - require relatively light compressive loads because their design tends to inflate or extend them into place with increased operating pressure.

Because of the high operating pressure, simplicity, possible economies and speed of opening and closing, a self-energizing seal is recommended. Such a seal would even be desirable for the compression gasket seal application for which the pretensioning could be greatly reduced.

A leakage of 5 lbs per day from the primary vessel seal is acceptable. This supports the use of a second seal and a weep channel to check the amount of leakage.

The seal material should be resilient, chemically resistant and stable at operating temperatures. It has been found that such ideal seals are not available and need to be further investigated and developed.

Because such materials and constructions are not available, and time to develop them is not permissible, a gasket of more conventional design is recommended.

The information following is directed toward a development program to integrate the engineering information available from vendors and from scientific sources into a very practical industrial approach to the problem.

Following are general factors considered in making a selection for suitable type of seal material and construction.

1. Complete sealing is ^{not} imperative. For added safety two seals in series might be desired with a weep channel between them. Some leakage is permissible.
2. The seal material must have sufficient strength in itself or provided with a pre-assembly construction to permit convenient removal and replacement under water with minimum tools or delay.
3. The seal material should be resilient to make high pretensioning pressures unnecessary. Self-energizing design is preferred.
4. The seal material should not stick to contact surfaces.
5. The seal preferably is in the lip of the vessel rather than in the cover in order that the heavy cover need not be inverted to replace a seal.
6. The seal is preferably well confined to reduce structural and physical limitations.
7. The seal material should preferably be readily available, economical and have long life
8. The seal material should preferably be continuous.
9. Repair or splicing of the seal material should be possible.
10. The seal material should have minimum susceptibility to radiation damage.
11. The seal material and any associated cell retaining material should be preferably non-contaminating to the system.

General requirements for the construction discussed include the following:

1. Temperature - 500° to 550°
Pressure - 600 to 660 psi Saturated steam
2. Type - Self-energizing - 2 seals in series (preferred)
3. Radiation at Gasket - Fast Neutrons 2×10^7 per centimeter square per second.
- Thermal Neutrons 1×10^8 per centimeter square per second.
- Gamma Rays - 7.6×10^8 per centimeter square per second.
4. The reactor will be in continuous operation for 3 to 4 months, at the end of which time the seals can be replaced. The seals are to be located in the lip of the high pressure vessel. Access to the seal is thru 18 feet of water necessitating tools for removal and replacement.
5. It is desired that recommendations be provided for self-energizing seals of single and multiple ring construction. Recommended dimensions and structural features should be indicated to permit inclusion in the overall design.

Because of the lack of structural strength and the relatively flimsy ring, it may be desirable to back up the construction with a metal ring integral or vulcanized to the seal material.

It is desired that quotations be provided in triplicate addressed to Koerper Engineering Associates for the different types of recommended structure and materials with and without backup rings,

5. if so feasible.

It is also requested that the material, which preferably should be aplicable, should also be quoted in costs per foot with some indication of vulcanizing techniques and necessary tools therefore.

Particularly desired are quotations on

1 - 3 - Recommended $3\frac{1}{2}$ in. O.D. Test seals specimen,

1 - 3 - Recommended 12 in. O.D. Test seals specimen,

1 - 10 - Recommended 90 in. O.D. Seals,

also tooling costs if any on above, and estimated delivery times of the various pieces.

In order to make reasonably valid comparison of the various sealing methods, recommendations should include such engineering information as seal cross section groove dimensions and tolerances, desired precompression and loading necessary to effect the seal.

E.C.K.

• RADIATION RESISTANCE OF PLASTICS
Reference Report - AEG-ORNL 928
"Physical Properties of Irradiated Plastics"

Report: KDM-AC-1011
Date: Aug. 28, 1959
Page: 46-A

<u>Plastics</u>	<u>Exposure</u> <u>10^{18} nvt</u>	<u>Change in Properties</u>
1. Mineral-filled furan and mineral-filled phenolics: Duralon, Haveg 41, asbestos-fiber Bakelite, asbestos-fabric Bakelite, and Karbate	10	Little change except for darkening in color.
2. Styrene polymers: Amphenol and Styron 411C	10	Little change except for darkening in color.
3. Modified styrene polymer: Styron 475	10	Impact strength and elongation decrease until the same as unmodified styrene polymers.
4. Aniline formaldehyde (Cibanite) and polyvinyl carbazole (Polelectron)	10	Tensile strength decreases a little.
5. Polyethylene and Nylon	10	Impact strength decreases but tensile strength increases. These plastics become so brittle that the corners of the specimens chip off.
6. Mineral-filled polyester: Plaskon Alkyd	10	Tensile strength and impact strength are decreased about 50%.
7. Unfilled polyesters: Selectron 5038 and CR-39	5	Develop small cracks. Tensile strength and impact strength decrease.
8. Phenolics with cellulosic fillers: paper-base Bakelite linen-fabric Bakelite, and Micarta	3	Become brittle, swell, and decrease in tensile and impact strength.
9. Melamine and urea: Melmac, Bettle, Plaskon urea, and Plaskon Melamine	2	Tensile strength and impact strength are decreased about 50%.
10. Unfilled phenolic: Catalin	1	Tensile strength and impact strength are decreased about 50%.

The characteristics of "semi-plastic" sealing material under simulated operating conditions should be determined as generally proposed in (S-1) - Page 44.

Extrusion characteristics will be important to meet fit tolerances for material under high temperature and pressure and should be included in the development. A retaining ring with extrusion sections of various widths could be easily used for this purpose.

The basic equipment shown below, is now in the laboratories of the A-C Nuclear Power Division. The accessory components recommended to make the test runs is relatively simple and inexpensive.

• SMALL RING SEAL TEST EQUIPMENT
for 3/4 In. O.D. Test Specimen

Report: KDM-AC-101
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NOTE SEAL CAN BE USED WITHOUT STEEL RING
TO DETERMINE CLAMPING SEALING PRESSURE.

⊗ THE CROSSSECTION OF THE SEAL
TO BE RECOMMENDED BY SUPPLIER.

ANY TEST SEAL RING WITH
SUITABLE RETENTION OR
WITH MECHANICAL RADIAL
PRESSURING. ⊗

PRESSURIZE WITH
HYDRAULIC JACK

• INNER EXPANDER RING - TEFLON SEAL

Report: KDM-AC-1011
Date: Dec. 11, 1958
Page: Sketch "H"-R - 49

SCALE = 8 TIMES

T.E. MANTSCH

B49

To adapt to Allis-Chalmers existing test equipment

NOTE #1:-

ITEM #5 TO BE REMACHINED TO 9.500 ±.010 DIA. TURN
OR MADE NEW

To establish flow
and extrusion
characteristics
of thermoplastic
seal materials

Proposed Seal Construction For Test

- A - Outer Ring - Spliced - Material: Aluminum or Stainless Steel.
- B - Seal - Solid Rings; Segmental; Spliced (Bonded)
Material - Teflon, Kel-F with Asbestos.
- C - Inner Ring - Solid - Material: Aluminum or Stainless Steel.

Features:

1. Preassembled unit - easily installed or removed.
2. Multi sealing surface.
3. Requires low initial compression.
4. Sealed medium supplies sealing force
5. Probably reusable.

• HIGH PRESSURE REACTOR VESSEL SEAL

Report: KDM-AC-1011

Date: Aug. 28, 1959

Page: 54

Proposed Seal Construction for Test

A - Outer Ring - Solid - Material: Aluminum or Stainless Steel

B - Seal - Solid Ring; Spliced (Bonded)

Material: Teflon, Kel-F with Asbestos

C - Inner Ring - Solid - Material: Aluminum or Stainless Steel

Features:

1. Preassembled unit - easily installed or removed.
2. Requires low initial compression.
3. Sealed medium supplies sealing force.
4. Probably reusable.

Tentative

Due to the limitations of non-metallic materials used for seals under moderate pressures and high temperatures, KEA is suggesting a novel self-energizing metallic seal that is activated by the internal pressure. The principle of this seal can be illustrated by a simple toggle, operating in the ten to zero degree range where the mechanical advantage would be from approximately 3 to 800 + respectively. Therefore, high sealing forces can be obtained at the lip of the seal with relatively low internal pressure.

This report should be considered as a preliminary analysis to establish the feasibility of this design. The calculations in this report are for a 12 in. dia., 3/4 in. height test ring that could be used in the present Allis-Chalmers' Test Cylinder.

In the following calculations a one inch section of the ring is used as a curved beam to determine the deflection under a given load. As long as the ends are free to move and the ratio of the section thickness to the radius is greater than ten to one, the straight beam deflection formula is used as a uniformly loaded beam. However, when the end movement is restricted, as is necessary to have a sealing condition, the beam section would react like an arc of an externally load ring.

In a relatively thin rectangular section cases arise where the flexural fiber stress is relatively so small that the resultant fiber stress is not different from the direct stress. Then the resultant is wholly tensile or compressive, and there is no neutral axis. (Ref. Mauer & Whitney - Strength of Materials - 2nd Ed. Art. 149).

• "DEFLECTITE" SEAL
Pressure Activated - Self-Energizing Seal
For Nuclear Reactor

Report: KDM-AC-1011
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Tentative

The elongation in the hoop direction is less than 15/100 of one percent at maximum deflection and, therefore, is not included in the calculations.

Test Ring Data:

Outside Diameter = 12.000 inches
Height of Section = .750 inches
Internal Pressure = 600 psi Max.

D = Deflection

H = Load (Total-Uniformly Distributed)

l = Beam Length = .625 inches.

E = Modulus of Elasticity - 28×10^6 for Stainless Steel
 10×10^6 for Aluminum

h = Height of Section - .028 (Stainless) & .040 (Aluminum)

I = Moment of Inertia

Z = Section Modules

S_o = Compressive Stress

D = $5WI^3/384EI$ (Free End Movement)

$I_s = h^3/12 = (.028)^3/12 = 18.3 \times 10^{-7}$

$I_A = h^3/12 = (.040)^3/12 = 5.33 \times 10^{-6}$

$Z_s = h^2/6 = (.028)^2/6 = 13 \times 10^{-5}$

$Z_A = h^2/6 = (.040)^2/6 = 2.67 \times 10^{-4}$

• "DEFLECTITE" SEAL
Pressure Activated - Self-Energizing Seal
For Nuclear Reactor

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Tentative

Test Ring Data - (Con't.)

Deflection for stainless steel

$$D = 5 \times (.625)^3 W/384 \times 28 \times 10^6 \times 18.3 \times 10^{-7} = .0000623 W$$

Deflection for Aluminum

$$D = 5 \times (.625)^3 W/384 \times 10 \times 10^6 \times 5.33 \times 10^{-6} = .0000594 W$$

Deflection at Maximum Pressure (Ends Free)

$$600 \times .75 \text{ sq.in.} \times .0000623 = .028 \text{ inches (Steel)}$$

$$600 \times .75 \text{ sq.in.} \times .0000594 = .0266 \text{ inches (Aluminum)}$$

Maximum Height Increase

$$38 \times 2\pi \times .916/360 - .5963 = .6075 - .5963 = .0112 \text{ inches.}$$

The force on the lip of the seal is very similar to that of a simple toggle, however, the initial pressure on the toggle is a concentrated load and equal to approximately one-half of the total uniform load in the seal application.

F = Applied Force

P = Lip Force = FC

a = Given Angle

C = Coefficient = $\cos a/2 \sin a$

$$\text{Initial Angle } a_1 = \tan a_1 = .0547/.313 = .1695 = 9^\circ-37'$$

$$\text{Max. Deflection Angle } a_2 = \tan a_2 = .0547 - .028/.313 = .0854 = 4^\circ-53'$$

$$\text{Coefficient C Angle } a_1 = .986/2 \times .167 = 2.95$$

$$\text{Coefficient C Angle } a_2 = .9964/2 \times .085 = 5.87$$

$$\text{PC } a_1 = .5 \times 450 \times 2.95 = 664 \text{ lb./Linear Inch}$$

$$\text{PC } a_2 = .5 \times 450 \times 5.87 = 1320 \text{ lb./Linear Inch}$$

• "DEFLECTITE" SEAL
Pressure Activated - Self-Energizing Seal
For Nuclear Reactor

Report: KDM-AC-1011
Date: Nov. 28, 1958
Page: 4 - 58

Tentative

It is difficult to make a complete paper analysis of the suggested test ring. The data and calculations in this report are to present the feasibility of this design.

A test program would be required to determine the most desirable shape and section profile. Several shapes are shown on the attached Sketch "A" that could be tested in the present Allis-Chalmers' 12 inch test cylinder. The most economical method of producing the test rings would be by spinning. The estimated tooling cost for both shapes would be approximately \$125.00 and a per piece charge of \$25.00. We are waiting for an accurate quotation from several spinning companies.

On Sketch "B" we have shown a proposed test fixture to accurately determine load, deflection and lip pressure under various conditions. The amount of spread of the ring section can easily be increased or decreased by the addition or removal of shims. The hydraulic circuit is shown on Sketch "C".

L. M. Koelsch

"DEFLECTITE" SEAL
• Pressure Activated - Self-Energizing Seal
For Nuclear Reactor

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Proposed Seat Tester

"DEFLECTITE" SEAL
Pressure Activated - Self-Energizing Seal
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Proposed Seat Tester - Hydraulic Circuit

• "DEFLECTITE" SEAL
Pressure Activated - Self-Energizing
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• "DEFLECTITE" SEAL
Pressure Activated - Self-Energizing
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• PRELIMINARY DESIGN DATA

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Design Data

1. Design pressure 660 psig
2. Design temperature 496°F
3. Normal operating pressure 600 psig
4. Hydrostatic test pressure 990 psig
5. Vessel wall thickness 3" total
 - a) Base metal SA 212-B 2-3/4"
 - b) Stainless clad 304 ELC 1/4"
6. Vessel shell I.D. 11'-0"
7. Minimum vessel closure opening 7'-6"
8. Max. heating & cooling rates 400°F/Hour
9. Weights of various components
 - a) Control rod drive mechanisms 8,500 lbs
 - b) Vessel cover Approx. 16,000 lbs
10. Normal shield pool temperature 100°F
11. Depth of shield pool water above parting line of closure 18 ft.
12. Shock load on vessel cover during control rod scram (force vertically downward) 80,000 lbs
13. Maximum permissible misalignment between control rods and guide tubes $\pm 1/32"$
14. Internal heat generation due to radiation at the inside surface of the cover. 400 BTU/HR-FT³

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Nuclear Power Division

JP:cw