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DEVELOPMENT OF RING-JOINT FLANGES FOR USE IN THE HRE-2

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OAK RIDGE NATIONAL LABORATORY
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ABSTRACT

Ring-joint flanges were studied extensively in thermal-cycle tests as part of the development work associated with design of Homogeneous Reactor Experiment No. 2 (HRE-2). The purpose of this program was to provide criteria for design, installation, and operation of joints that would remain leak-tight under reactor operating temperatures and pressures. Joints ranging from 1/2 in., 1500 lb to 4 in., 2500 lb and with various initial bolt loadings were cycled between room temperature and 636°F. It was demonstrated that when joints are made up to HRE-2 standards and specifications, leak rates of less than 0.25×10^{-3} g of water per day per inch of gasket pitch diameter can be routinely attained. Undamaged gaskets may be reinstalled or new gaskets may be used with equal probability of achieving acceptable leak rates. The system installed in HRE-2 was provided with a high-pressure buffer system to ensure that the small amount of leakage to the cell would be nonradioactive.

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1. Summary and Conclusions

As part of the development work carried out in conjunction with the design of the HRE-2, a number of flange development tests were conducted to study the characteristics of ring-joint flanges during thermal-cycling operation. It was hoped that information from this program would provide criteria for design, installation, and operation of joints which would remain leaktight over the range of reactor operating temperatures and pressures. Joints ranging in size from 1/2 in., 1500 lb to 4 in., 2500 lb were cycled between room temperature and 636°F under various initial bolt loadings. Leak rates were observed. It was demonstrated that when joints are made up to HRE-2 standards and specifications, leak rates of less than 0.25×10^{-3} g of water per day per inch of gasket pitch diameter can be routinely attained. Undamaged gaskets may be reinstalled or new gaskets may be used with equal probability of achieving acceptable leak rates. The flanges installed in the reactor were provided with a high-pressure buffer system (see sec. 2.2.1) to ensure that the small amount of leakage to the cell would be nonradioactive.

Flange test No. 1 (see sec. 2.3) demonstrated that making up ring-joint flanges that remain leaktight during thermal cycling was a problem which required additional development work. The test was set up to compare the performance of 1500- and 2500-lb flanges, but it was impossible to keep either flange leaktight during thermal cycles. Thermal profiles for the 1500-lb flange and temperature differences for bolt and gasket with the flange insulated and uninsulated were determined during the test.

Furnace tests (see sec. 5.1) were conducted to demonstrate the effect of added resiliency in the bolting system in reducing the stress induced by differential thermal expansion. The increased resiliency was effective, but the total required resiliency could not be attained by reducing the bolt shank diameter alone. Thus a supplementary mechanism was necessitated,

and ferrules were ultimately used for this purpose. The ferrule length, however, was not optimized in this investigation or those which followed.

Flange test No. 2 (see sec. 2.4) was the main effort of the flange development program, and during this test data were accumulated on the flanges specified (see sec. 3) for the HRE-2. It was determined that, for the flange to remain tight indefinitely in service, the bolts should be tightened initially to a load about twice that recommended by the ASME code. The test demonstrated that flanges under simulated reactor service experienced a leak rate of less than 0.25×10^{-3} g of water per day per inch of gasket pitch diameter. It also showed that rings having a pitch diametral mismatch approaching that allowed by ASA specifications (but having surface finish and seating-angle tolerances to HRE-2 specifications) would experience about the same leak rate. Further tests, with and without ferrules, showed that the effectiveness of the ferrules was acceptably close to the calculated values. It was also shown that the load that remains after cycling in a ring joint of the design used is limited by the gasket stress.

Torque-load tests (see sec. 5.2) were conducted to determine the proper torques to use in stressing reactor flange bolts. These tests demonstrated that the bolts of a given lot will exhibit similar torque-load ratios, but that different lots will behave differently.

The conclusions drawn from the entire program may be summarized:

1. Flanges made up to HRE-2 specifications (see sec. 3) may be expected to experience initial leak rates of less than 1×10^{-3} g of water per day per inch of gasket pitch diameter, dropping to less than 0.25×10^{-3} g of water per day after several cycles. This leak rate, when compensated for by the use of a buffer system (see sec. 2.2), is considered acceptably low for flanges in aqueous homogeneous reactor service.

2. Undamaged gaskets may be reinstalled or new gaskets may be used with equal likelihood of achieving an acceptable leak rate.

3. Flanges may be insulated to reduce heat loss to the reactor cell, without incurring excessive leak rates during thermal cycling.

4. Closer tolerances on the pitch diameter of the flange groove and of the gasket and on the contact angle of the two and finer control of the

surface finish of the contacting surfaces seem to enhance the likelihood of satisfactory fitup and the achievement of leaktightness. Until further tests establish a definitive relationship between tolerance and leak rate or between surface finish and leak rate or both, it is recommended that specifications for reactor flanges require the closest tolerances that have been met without excessive difficulty by flange and gasket manufacturers. The specifications given in Section 3 have been successfully met.

5. The flange should be as hard as possible and the gasket should be as soft as possible to minimize the deformation of the flange groove with successive reassemblies and consequently to extend the service life of the flange. Limiting hardnesses which have been successfully met by flange and gasket manufacturers are given in Section 3.

6. Care must be taken during manufacture, handling, storage, and installation to protect the seating surfaces of the flange grooves and of the gaskets from damage. This is perhaps an obvious precaution, but it is certainly difficult to put into practice.

7. Bolts should be tightened to an initial stress about twice the code recommended value.

8. There is no basis for choosing between oval and octagonal cross-section gaskets as a result of the present program, but the octagonal gasket is preferred because it lends itself to more precise machining and inspection and because it seats with less plastic deformation.

9. The use of ferrules and reduced-shank bolts lowers the stresses induced by differential thermal expansion and thus enhances the probability of a joint remaining leaktight through many thermal cycles.

10. The maximum bolting loads which may be effectively applied to the joint are believed to be limited by a maximum gasket compressive stress of about 32 000 psi (based on the ring cross-sectional area parallel to the face of the flange) or a maximum gasket contact stress of about 160 000 psi.

11. Since the maximum bolting loads which can be effectively applied to a flange are limited by the gasket loads, it may be concluded that torque wrenches are adequately safe for field control of bolt tightening.

It is desirable, however, to determine the torque-to-load ratio for each lot of bolts to effect uniform bolt loading and thereby to promote leak-tightness. The torque-load calibration can be made readily in the field.

12. Both 2500- and 1500-lb flanges appeared to be adequate over the range of the tests.

13. Editor's note: As of March 1961, following 2 1/2 years of service, the HRE-2 flanges were still functioning properly. Not one instance of excessive leakage through a flange had occurred, as determined by monitoring the buffer zone between the two ring seals.[1]

2. Flange Tests

2.1. Objective and Scope

The objective of the flange development program was to develop criteria such that a mechanical pipe joint could be made up that would exhibit, initially and through thermal cycles, a small enough leak rate to be acceptable for aqueous homogeneous nuclear reactor service. Numerous thermal cycles of random length were made from 70 to 636°F (0- to 2000-psi saturated-steam pressure) at a maximum rate of ~100°F per hour.

The work reported here was limited to flanged pipe joints of the type specified for the HRE-2, consisting of stainless steel welding-neck flanges in the size range 1/2 to 4 in., 1500- and 2500-lb classes, with stainless steel ring-joint gaskets and carbon steel bolts. Since prior experience with this type of joint had been somewhat erratic, the program was directed at developing refinements which would make the joint more reliable. Detailed descriptions of the components of the joint are given in Section 3.

In conjunction with various phases of the flange development program, a literature search was conducted and pertinent reports were reviewed. These reports are listed chronologically in the Bibliography.

[1] J. A. Watts, Operating Experience with the HRT Flange Leak Detector System, from 12-6-57 Through 2-11-59, ORNL CF-59-5-116, May 22, 1959.

2.2. Discussion of the Joint Studies

Some features of the HRE-2 joints (Fig. 1) deviate from conventional usage of ring-type joints. In order to ensure against leakage of radioactive material from the reactor system into the cell at the flanged joints and to provide for detection of leakage, the annular space between the inner and outer sealing surfaces of the gasket is pressurized with heavy water at a pressure greater than that in the reactor system. If a leak should develop across the inner seal, heavy water will leak into the reactor system instead of radioactive material leaking outward to the leak-detector system. The presence of such a leak is indicated by the pressure drop occurring in the heavy-water system. Thus it is necessary for the gasket to seal against both the inner and the outer surfaces of each flange groove. In conventional usage of ring-type joints, such a double seal is not required.

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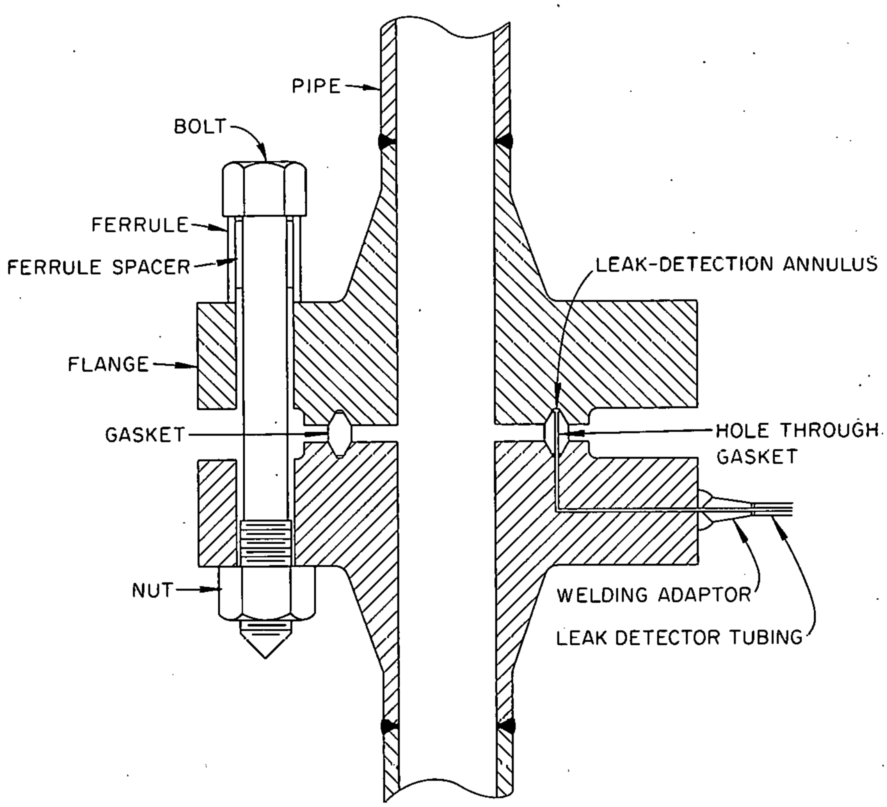

Fig. 1. HRE-2 Flanged-Joint Assembly.

The flanges, as well as the piping, are insulated to avoid excessive heat loss to the cell. All flanges in the process system are made of stainless steel and clamped with carbon steel bolts. Differential thermal expansion of the different materials and the existing temperature differences can cause plastic deformation in the gaskets when the joints are cycled through their normal temperature range. Thus there is a tendency to loosen the joint and cause leakage. This problem is partly alleviated by introducing greater resilience into the flange-bolt system to permit the system to absorb differential expansion elastically with less induced stress in the gasket. Greater resilience is introduced by using long bolts with pipe ferrules and by machining the shank of the bolt to a smaller diameter. Other methods, not used in the present investigation, are the placing of Belleville washers under the head of the bolt and counterboring of the flanges for ferrules to reduce the effective length over which differential expansion would occur.

In HRE-2 service, 2500-lb flanges are used at 2000 psi and 600°F. Rating of these flanges at 600°F is 4620 psi. Flanges of the 1500-lb class are rated at 2770 psi and therefore would be adequate for this service. The 2500-lb design was selected because of its greater rigidity, although the thermal expansion stresses are slightly greater.

The hardness of the stainless steel used in the flanges is essentially equal to that of the rings, whereas it is preferable to use rings of material much softer than the flanges. To produce a leaktight seal between the ring and the groove, it is necessary for plastic deformation to occur in the metal to seal off fine surface irregularities at the interface. If, owing to the hardness of the ring and the use of excessive bolting loads, the flange grooves are appreciably deformed, it may be difficult to seal the joint after reassembly. Therefore, it is essential in HRE-2 to exercise care to attain a good surface finish on the rings and grooves, to protect this surface from any mars or scratches, and to limit bolt loads to the lowest values that will effect an adequate seal.

The joints will have to be broken and remade from a distance. This intensifies the problem of making a careful inspection of the flange grooves before the reassembly of a joint and greatly increases the difficulty of surface repair.

2.3. Flange Test No. 1

Flange test loop No. 1 consisted of one pair of 4-in., 2500-lb and one pair of 4-in., 1500-lb stainless steel ring-joint welding-neck flanges and was used for preliminary studies and for the development of instrumentation to be used in flange test loop No. 2. The system was filled with water to a predetermined level and heated to 636°F (2000 psi) by an electric heater. Cooling was accomplished by blowdown of the system contents. Approximately four cycles per day could be achieved in this manner.

The initial test of the flanges was made with gold-plated gaskets. The lowest bolt load that would produce an initial seal was used on the theory that with low bolt loading, deformation of the gasket and the ring groove would be low and the joint would have the best chance of a long service life. Leakage was experienced with the first thermal cycle and continued through successive increases in bolt loading. Even after the original gaskets were replaced with unplated gaskets, it was never possible to tighten the flanges so that they would remain leaktight through a significant number of thermal cycles. (The bolt loadings used in these tests did not approach those finally recommended for HRE-2 use; also, the rings and grooves were standard commercial products.)

Bolt loads were measured as the system was hydrostatically pressurized to 3000 psi, and the bolt, pressure, and gasket loads for the two flanges are plotted against pressure in Fig. 2. Since the available gasket load is a measure of the ability of the joint to resist leakage, the curves indicate that both the 1500- and 2500-lb flanges appear capable of loading the gasket sufficiently in the pressure range tested.

In spite of the inability to secure leaktight joints, it was decided to run some additional cycles to determine the temperature profiles in the flanges during thermal cycling and to observe the effect of insulating the flanges on these profiles. These data were to be used to help determine whether the flanges in HRE-2 should be insulated.

Figure 3 is a plot of pipe temperature and gasket-bolt temperature difference against a time base for the four cycles, two with insulated flanges and two with uninsulated flanges. It may be observed that there is little difference in the maximum gasket-bolt temperature difference for the four cycles, although the equilibrium temperature difference is greater for the uninsulated cases. Heating rate does not seem to affect the maximum difference.

Equilibrium temperature profiles determined during these tests are presented in Fig. 4 for the insulated and uninsulated conditions of the 4-in., 1500-lb

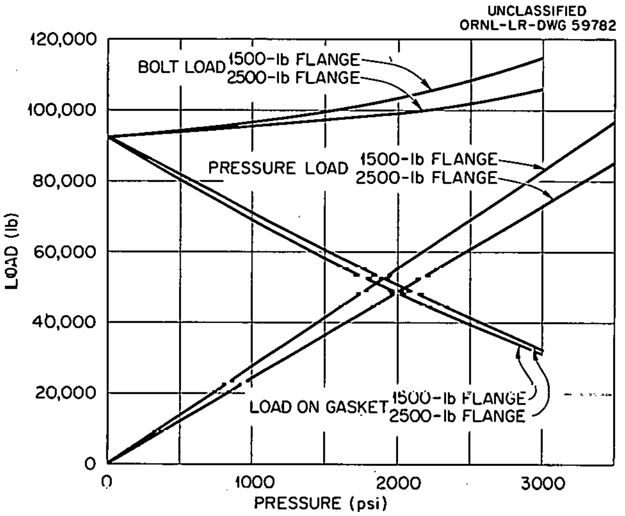

Fig. 2. Effect of Hydrostatic Pressure on Bolt and Gasket Loads.

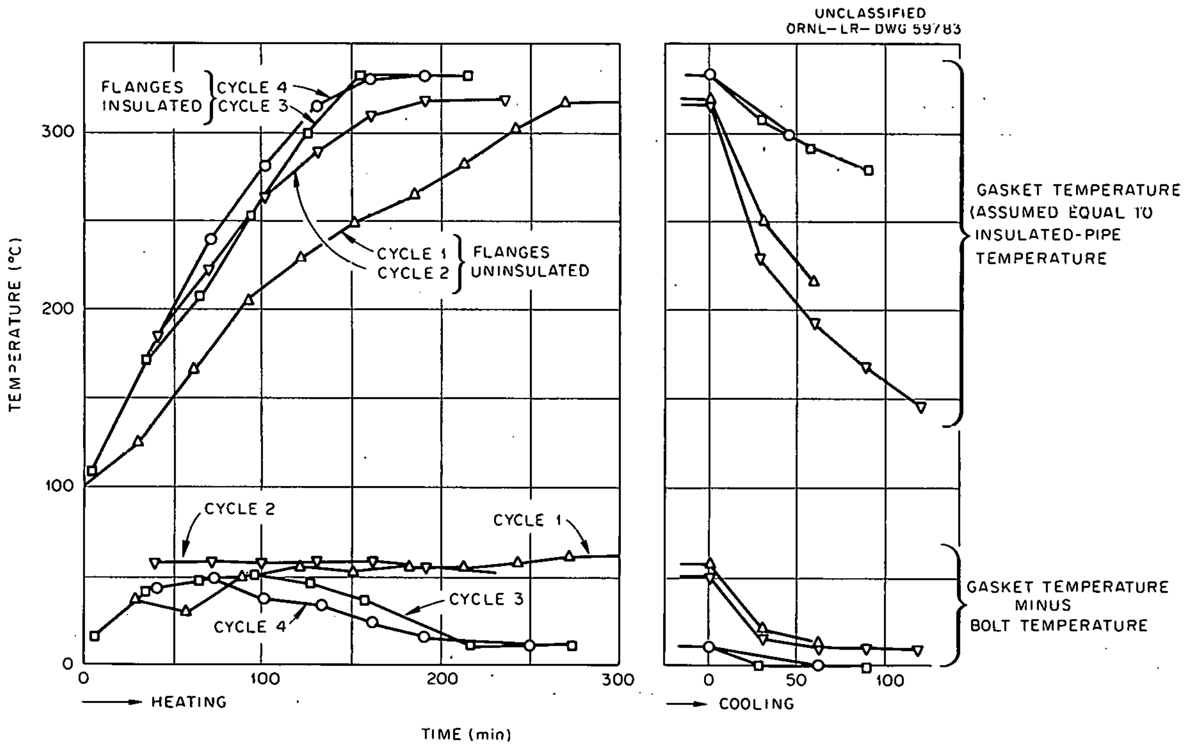

Fig. 3. Temperatures During Thermal Cycling.

flange. An attempt was made to vary the heating rates for each pair of otherwise identical cycles, but the available boiler capacity prevented any significant variation in the heating rates for the uninsulated flanges.

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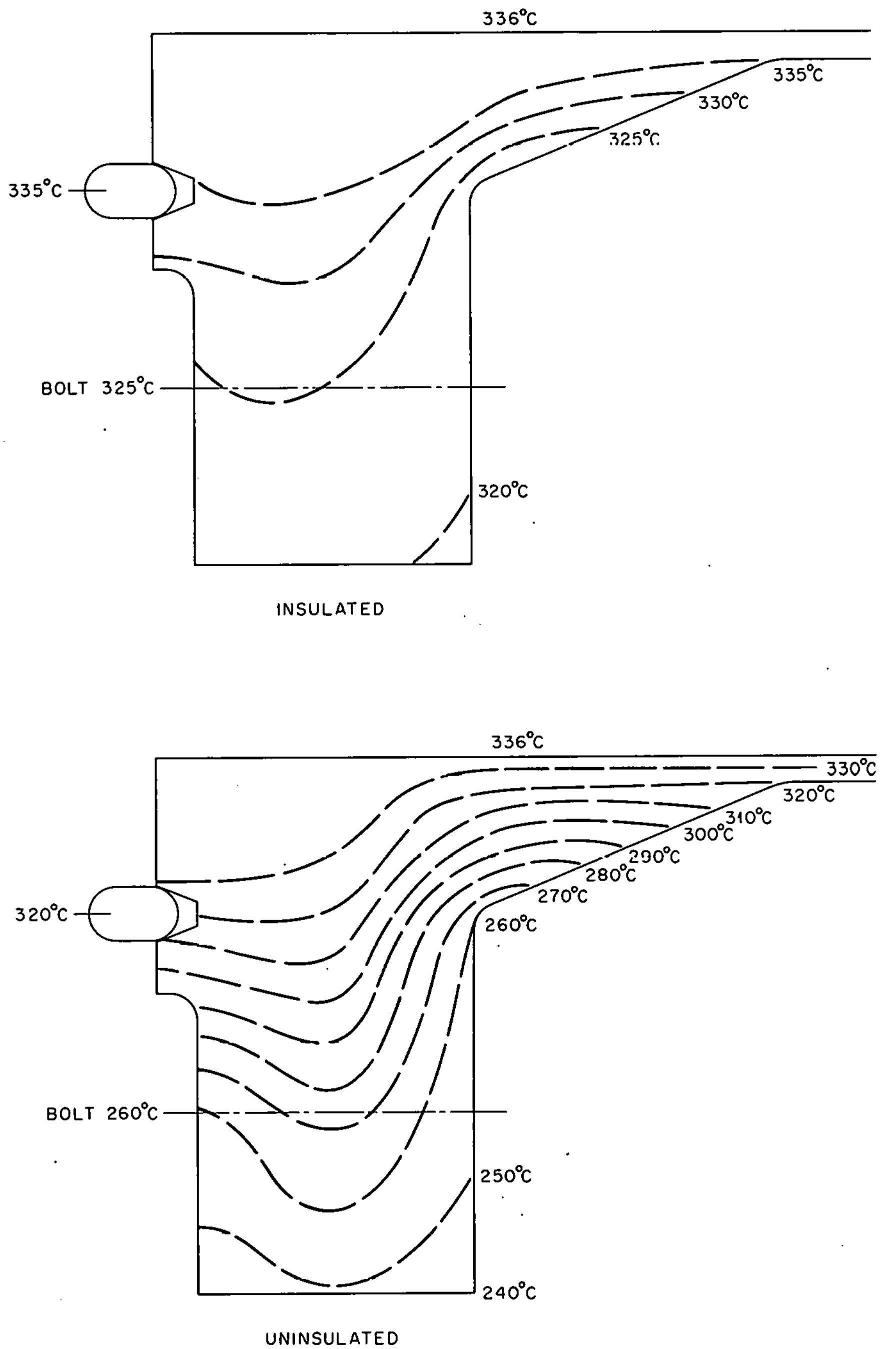


Fig. 4. Steady-State Thermal Gradients in 4-in. 1500-lb Flange.

2.4. Flange Test No. 2

A multistation flange test facility (Fig. 5) was constructed to serve as a general-purpose test facility for the investigation of piping connectors for high-temperature aqueous service. The facility consisted of a natural-convection thermal-cycling loop heated by steam from a 40-kw electric boiler and was capable of cycling its eight test stations at a rate of 100°F per hour from 17 psia at 220°F to 2000 psia at 636°F. The facility was instrumented for automatic cycling, and all data, except pin extensometer readings, could be read at the control panel.

The facility was initially operated to test the flanged joints which had been specified (see sec. 3) for HRE-2. These joints were specified, somewhat intuitively, on the basis of available experimental data and experience. Early operation of the facility (through cycle 43) was clouded by equipment and instrumentation difficulties. During this period the instrument systems were improved and calibrated, and the equipment was altered to achieve reliable service.

In early runs it was attempted to produce tight sealing with initial bolt stresses in the range of 20 000 to 30 000 psi. It was soon apparent that bolt loading was reduced following thermal cycling so that retightening twice was necessary for a joint to remain leaktight.

It was finally demonstrated in run A (cycles 44 to 52) that by initially tightening the bolts to a bolt stress of about 45 000 psi, asymptotic bolt loading of about 30 000 psi could be obtained following several thermal cycles. Figure 6 illustrates the variation in bolt loading as a typical 3 1/2-in., 2500-lb flange was cycled; the bolt stress at room temperature was observed in flange test No. 2, while the stress shown in the figure at operating temperature was calculated. The method of initially overstressing the bolts is recommended for reactor service.

It was hoped also in run A to compare oval and octagonal ring gaskets. No significant difference in asymptotic bolt loading (Table 1) or leakage (Table 2) could be observed between the two types of gaskets. The flange grooves showed slight, but essentially equal, deformation regardless of the type of ring used. The observed deformation of the oval rings was

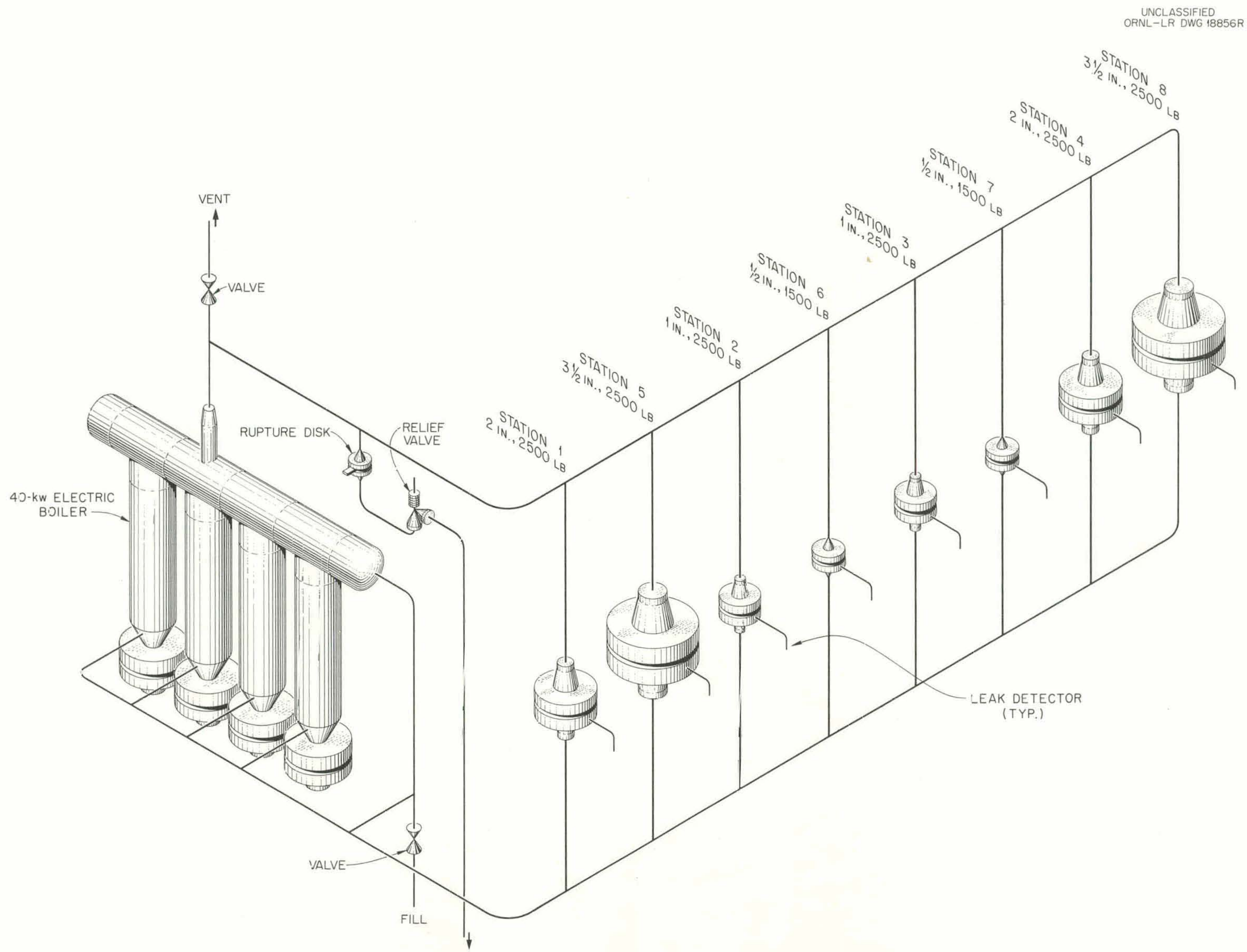


Fig. 5. Flow Diagram of Flange Test Facility.

such as to produce an essentially octagonal ring after several thermal cycles; no deformation of the octagonal rings was noticed.

Runs A through I were intended to demonstrate that the HRE-2 joints were essentially leak free. During this period one set of rings was installed and reinstalled a total of five times (for each flange pair); then four new sets of rings were installed (per flange pair). Leak rates measured at maximum temperature (636°F) and pressure (2000 psi) are summarized in Table 2. It may be observed that the leak rates were less than 1×10^{-3} g of water per day per inch of gasket pitch diameter after the first cycle of a run and continued to decrease with additional cycles to less than 0.25×10^{-3} g per day. There is no significant difference between the leak rates measured with new or with reinstalled gaskets.

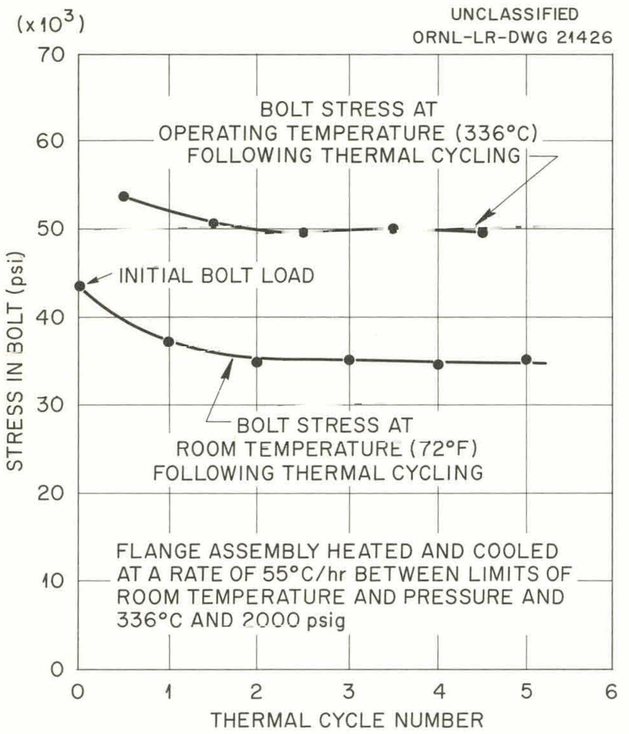


Fig. 6. Variation in Bolt Loading During Thermal Cycling of a Typical 3 1/2-in. 2500-lb Flange.

Table 1. Effect of Initial Bolt Stress on Asymptotic Bolt Stress After Thermal Cycling[a]

Station	Flange Size		Initial Stress (psi)		Stress Reduction (%)		Apparent Asymptotic Residual Stress (psi)		Type of Ring
	in.	lb	Prior to Cycle 44	Cycle 44	Prior to Cycle 44	Cycles 44-48	Prior to Cycle 44	Cycles 44-48	
1	2	2500	26 000	41 000	41	34	16 000	27 000	Oval
2	1	2500	23 000	36 000	20	41	18 000	21 000	Octagonal
3	1	2500	23 000	36 000	27	41	17 000	21 000	Octagonal
4	2	2500	26 000	41 000	34	32	16 000	28 000	Oval
5	3 1/2	2500	28 000	42 000		24		32 000	Octagonal
6	1/2	1500	17 000	38 000	40	32	10 000	26 000	Oval
7	1/2	1500	17 000	38 000	24	33	13 000	25 000	Oval
8	3 1/2	2500	28 000	42 000	43	32	16 000	28 000	Octagonal

[a]All stresses are based on reduced shank area.

Table 2. Flange Leakage During Thermal Cycling

Run	Date (1957)	Cycle No.	Number of Cycles In Run	Leak Rate Measured at 330°C (g of water per day)								Comments
				1/2-in., 1500-lb Flange		1-in., 2500-lb Flange		2-in., 2500-lb Flange		3 1/2-in., 2500-lb Flange		
				No. 6	No. 7	No. 2	No. 3	No. 1	No. 4	No. 5	No. 8	
				$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	
A	3-19	44	1	2.5	1.32	0.93	1.27	4.0	6.4	7.4	6.6	New rings, HRT specification
	20	46	3	1.33	0.46	0.69	0.94	1.94	4.7	5.0	4.6	
	4-1	52	9	1.1	0.62	0.54	0.73	1.4	3.15	3.4	2.7	
B	4-4	53	1	1.44	0.65	0.69	0.85	2.0	2.6	2.85	4.04	Reinstalled rings, HRT specification
	5	55	3	1.15	0.54	0.55	0.74	1.08	2.74	2.87	3.0	
	6	57	5	0.68	0.46	0.46	0.74	1.06	2.58	3.2	2.73	
	8	61	9	0.72	0.44	0.46	0.59	1.08	2.55	2.45	2.36	
C	4-11	62	1	1.0	0.51	0.47	0.59	0.97	1.75	2.05	3.25	Reinstalled rings, HRT specification
	12	63	2	0.70	0.34	0.36	0.47	0.83	1.5	1.9	2.35	
	13	64	3	0.60	0.40	0.4	0.54	1.16	1.6	2.04		
	14	65	4	0.61	0.39	0.39	0.53	1.0	1.6	1.95	2.6	
	15	66	5	0.26	0.35	0.37	0.46	0.83	1.3	1.55	1.9	
D	4-18	67	1	0.3	0.51	0.62	0.66	1.55	1.75	2.2	14.5	Reinstalled rings, HRT specification
	19	67	1	0.36	0.34	0.39	0.52	0.86	1.4	2.1	2.4	
	22	67	1	0.23	0.22	0.27	0.32	0.6	1.08	1.3	1.7	
	23	68	2	0.54	0.2	0.25	0.28	0.51	1.06	1.2	1.1	
	24	70	4	0.27	0.14	0.18	0.20	0.40	0.82	0.89	1.6	
	25	72	6	0.41	0.22	0.26	0.3	0.59	1.08	1.3	2.0	
	26	74	8	0.42	0.21	0.25	0.33	0.55	1.27	1.3	2.4	
	27	76	10	0.35	0.19	0.22	0.27	0.5	1.27	1.2	2.3	
	28	78	12	0.34	0.2	0.23	0.28	0.5	1.3	1.3	2.4	
	29	80	14	0.33	0.21	0.23	0.28	0.51	1.32	1.26	2.5	
E	5-2	82	1	0.66	0.43	0.38	0.44	1.3	1.5	5.6	1.1	Reinstalled rings, HRT specification
	3	84	3	0.42	0.32	0.75	0.37	0.9	1.15	2.9	2.3	
	6	89	8	0.31	0.24	0.25	0.30	0.78	1.4	2.2	1.8	
F	5-16	91	1	1.1	1.5	1.15	1.2	4.1	3.4	7.8	16.0	New rings, HRT specification
	17	93	3	0.75	0.76	0.64	0.8	1.7	2.9	4.0	6.4	
	20	99	9	0.41	0.43	0.44	0.52	1.08	3.1	2.8	3.3	

Table 2. (Continued)

Run	Date (1957)	Cycle No.	Number of Cycles In Run	Leak Rate Measured at 330°C (g of water per day)								Comments
				1/2-in., 1500-lb Flange		1-in., 2500-lb Flange		2-in., 2500-lb Flange		3 1/2-in., 2500-lb Flange		
				No. 6	No. 7	No. 2	No. 3	No. 1	No. 4	No. 5	No. 8	
				× 10^{-3}	× 10^{-3}	× 10^{-3}	× 10^{-3}	× 10^{-3}	× 10^{-3}	× 10^{-3}	× 10^{-3}	
G	5-23	101	1	1.7	1.2	1.0	1.25	4.1	4.3	9.8	7.9	New rings, HRT specification
	24	103	3	0.8	0.63	0.59	0.72	1.6	2.0	3.8	4.0	
	27	108	8	0.47	0.37	0.44	0.47	1.03	1.5	2.6	2.9	
H	5-31	112	4	0.59	0.49			1.1	1.6	3.6	3.2	New rings, HRT specification
	6-3	118	10	0.43	0.37	0.41	0.41	0.91	1.2	2.6	2.5	
I	6-10	125	7	0.51	0.47	0.32	0.51	1.7	1.9	2.6	2.7	New rings, HRT specification
J	7-1	126	1	0.81	0.67	0.52	0.61	1.2	2.5	3.1	2.7	Mismatched rings
	2	128	3	0.51	0.41	0.41	0.50	0.73	1.4	2.3	2.2	
	3	130	5	0.42	0.38	0.36	0.47	0.61	1.3	2.1	3.1	
	5	134	9	0.37	0.26	0.31	0.37	0.49	1.2	1.6	2.2	
	8	140	15	0.35	0.26	0.31	0.36	0.50	2.1			
K	7-17	141	1	0.30	0.30	0.36	0.42	0.51	0.92	1.7	2.7	Mismatched rings reused
	18	143	3	0.34	0.28			0.43	1.0	1.7	2.3	
	19	145	5	0.33	0.27	0.30	0.40	0.51	2.5			
	22	151	11	0.33	0.28	0.32	0.41	0.51	11.0	1.7	0.87	
	23	153	13	0.32	0.30	0.10	0.36	0.44	14.0			
	24	155	15	0.28	0.24	0.26	0.37	0.62	17.0	1.4	2.2	
	25	157	17	0.29	0.26	0.28	0.37	0.39	21.0	1.6	2.1	
	26	159	19	0.26	0.25	0.26	0.37	0.38	22.0	1.6	1.8	
	29	165	25	0.28	0.25	0.41	0.29	0.41	31.0	1.6	1.8	
L	10-15	200	1		0.18	0.16[a]	0.27					Test of ferrules
	16	202	3		0.13	0.18[a]	0.23					
	17	204	5		0.14	0.14[a]	0.21					
	18	206	7		0.13	0.16[a]	0.20					
	21	212	13		0.11	0.16[a]	0.20					
M	12-16	226	1		4.26[a]		0.49[a]		1.69[a]	3.49[a]	7.24	Test of ferrules
	17	228	3		4.58[a]		0.27[a]		0.96[a]	1.93[a]	7.24	
	18	230	5		4.74[a]		0.09[a]		1.37[a]	1.41[a]	4.74	
	19	232	7		3.22[a]		0.31[a]		1.20[a]	1.29[a]	1.97	
	20	234	9		8.04[a]		0.18[a]		0.79[a]	0.18[a]	2.81	

[a] In these runs, the bolting system did not contain ferrules.

Runs J and K were devoted to the study of leak rates with flange grooves and gaskets having pitch-diameter mismatch greater than was permitted by the HRE-2 specifications and more nearly approaching the mismatch permitted by ASA B16.20-1956.[2] Here again, there was no significant difference in leakage experience with new or with reinstalled gaskets.

In these tests the flanges were assembled twice with rings having pitch diameters approximately 10 mils larger than the pitch diameters of the flange grooves, but the closer tolerance on the seating angle and the surface finish was retained. Upon disassembly, it was observed that the pitch diameter of the rings had been reduced to a diameter very close to that of the flange grooves. When the imposed deformation and the residual deformation (summarized in Table 3) are plotted on a compressive stress-strain diagram for stainless steel, they indicate that a ring crushed to the contour of the flange groove would exhibit a residual deformation very close to that measured. It was also observed that the bolts retained less of their initial loading with the mismatched gaskets than with the close-tolerance gaskets.

Leak rates at temperature and pressure for this series (runs J and K) are summarized in Table 2. With the exception of one group of data for a 2-in., 2500-lb flange, there is no significant difference between the leak rates measured for mismatched gaskets and close-tolerance gaskets. The exception was a 2-in. flange which exhibited a leak rate that increased with cycling. No defect was observed in the gasket on disassembly, or is there any explanation for this inconsistent behavior.

The general conclusion drawn from these tests is that in a radioactive environment it is better to procure gaskets to close tolerance than to depend on in-place forging. The better initial fitup achieved by controlling pitch diameter, surface finish, and seating-angle tolerances results in more consistent performance of joint components and greater reliability in reactor service.

Runs L and M were devoted to a study of the effectiveness of ferrules in reducing the thermal bolt stress induced by differential expansion of

[2]American Standard ASA B16.20 (1956), Ring Joint Gaskets and Grooves for Steel Pipe Flanges.

Table 3. Deformation of Mismatched Gaskets After Thermal Cycling

	Flange[a]										
	6T	7T	2T	2B	3T	1T	1B	4T	5T	5B	8T
Groove pitch diameter, in.	1.5625	1.5625	2.375	2.375	2.375	4.0	4.0	4.0	6.1875	6.1875	6.1875
Initial ring mismatch, mils	9.7	9.7	10.1	10.1	10.1	10.1	9.7	10.6	10.3	10.8	10.8
Change in ring diameter after cycling, mils	8.3	6.4	7.2	7.8	5.5	8.6	8.9	8.9	6.3	6.9	6.3
Imposed deformation, in./in.	0.0062	0.0062	0.0043	0.0043	0.0043	0.0025	0.0024	0.0027	0.0017	0.0017	0.0017
Plastic deformation, in./in.	0.0053	0.0041	0.0030	0.0033	0.0023	0.0022	0.0022	0.0022	0.0010	0.0011	0.0010

[a]T = top; B = bottom.

carbon steel bolts and the stainless steel flanges. In this test the thermal-load range, as evidenced by pin extensometer (see sec. 4.1) readings taken at the top and at the bottom of the cycles, was measured and was compared with theoretical values of thermal load (see Table 4). The tabulated experimental data are qualitative only; they were not corrected for thermal calibration (see sec. 4.1). The thermal correction is needed because of the suspected nonisothermal condition of the bolts, but the experimental data obtained to make the correction were insufficient. It should be noted that the measured values closely approach the calculated ones for smaller flanges, which theoretically approach thermal equilibrium more rapidly because of their lower heat capacity and smaller dimensions. In all cases the measured thermal-load range is smaller when ferrules are used on the flange bolts.

Cycles 240 through 248 were devoted to a thermal calibration of the pin extensometers (described in sec. 4.1) and resulted in some useful information on flange-bolt-gasket action. These data are presented in Table 5. During these cycles the flanged joints were made up and were cycled four times (to bring the bolts near their asymptotic load); then, with the assembly at pressure and temperature, one bolt was removed and its indicated change in elongation between the heated and room-temperature conditions was observed. From the data in Table 5 it is evident that the load induced in the joint by the differential expansion of the stainless

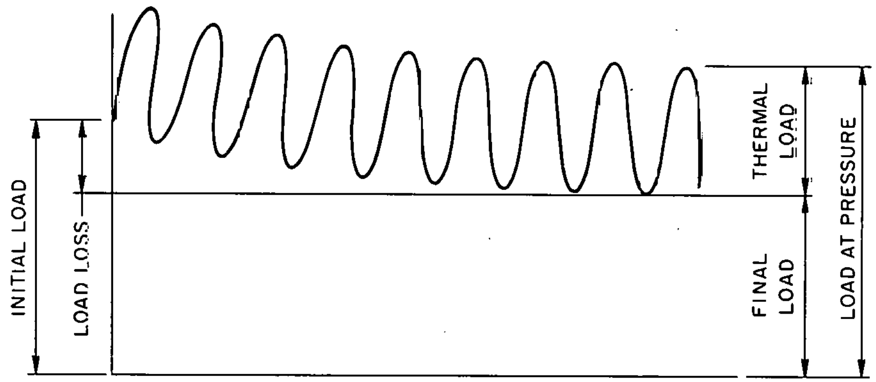
Table 4. Comparison of Thermal Loading of Bolts with and Without Ferrules (Runs L and M)

	Flange Size and Number						
	1/2 in., 1500 lb, No. 7	1 in., 2500 lb		2 in., 2500 lb		3 1/2 in., 2500 lb	
		No. 2	No. 3	No. 1	No. 4	No. 5	No. 8
With Ferrule							
Measured thermal load, lb	15 600	22 600	31 800	59 500			54 500
Calculated thermal load, lb	16 100	30 400	30 100	70 600			153 700
Ratio of measured to calculated load	0.97	0.75	1.00	0.84			0.35
Without Ferrule							
Measured thermal load, lb	15 800	37 000	39 300		78 800	100 900	
Calculated thermal load, lb	19 700	38 500	38 500		89 100	193 000	
Ratio of measured to calculated load	0.80	0.96	1.02		0.88	0.52	

Table 5. Comparison of Theoretical and Actual Thermal Loads (Cycles 240 to 248)

	Flange Size and Number						
	1/2-in., 1500-lb,	1-in. 2500-lb		2-in., 2500-lb		3 1/2-in., 2500 lb	
	No. 7	No. 2	No. 3	No. 1	No. 4	No. 5	No. 8
Ferrules	No	Yes	No	Yes	No	No	Yes
Initial load per flange, lb	32 300	57 300	51 000	169 300	160 000	330 300	378 600
Asymptotic load at pressure, lb	43 000	75 500	78 000	231 000	238 000	436 000	453 600
Measured thermal load, lb		33 200		110 500			156 600
Theoretical thermal load, lb	19 700	30 000	38 500	70 000	89 100	193 000	153 700
Final load (after completion of cycles 240-248), lb		42 300		120 500			297 000
Load loss, lb		15 000		43 000			81 600
Hydrostatic load, lb	2 400	6 700	6 700	19 900	19 900	48 600	48 600
Load on gasket at pressure, lb	40 600	68 800	71 300	211 100	218 100	387 400	405 000
Asymptotic gasket stress[a]							
Midplane area, psi	26 400	29 500	30 500	38 400	39 800	32 000	33 300
Contact area, psi	163 600	182 400	129 000	167 200	193 200	128 100	133 500

[a]Based on nominal dimensions.



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steel flange and the carbon steel bolting is very close to the calculated value. (In the calculation the flange is assumed to be infinitely rigid, since the reduction in load that may be expected from flexibility of the flange is a very small amount.) It is also evident that an appreciable percentage of the imposed load (initial load + thermal load) is lost during the cycling. Since the gasket stresses (contact stress and compressive stress) corresponding to the asymptotic bolt load (at temperature, measured when the bolt was removed) is almost constant for all sizes and conditions of flanges, it may be surmised that the maximum load the joint can withstand is limited by the gasket stresses. This surmise could be established by additional testing at higher initial load.

3. The HRE-2 Flanged Joint

3.1. Assembly

The bolts on the HRE-2 flanged joints are tightened as indicated in Table 6, with every precaution being taken to protect the gasket and ring-groove surfaces from damage during the process. The bolts are tensioned, with a torque wrench, to approximately the torques given in Table 6, the bolts being tightened in increments of the specified torque and in a

Table 6. Bolt-Tightening Specifications for HRE-2

Flange Size		Bolting		Ring Size	Approximate Torque (ft-lb)	Total Bolt Load (lb)
in.	lb	No.	Size (in.)			
1/2	1500	4	3/4	R 12	115	52 600
1/2	2500	4	3/4	R 13	115	52 600
3/4	2500	4	3/4	R 16	115	52 600
1	1500	4	7/8	R 16	190	71 300
1	2500	4	7/8	R 18	190	71 300
2	2500	8	1	R 26	250	185 600
2 1/2	2500	8	1 1/8	R 28	400	229 100
3	2500	8	1 1/4	R 32	700	282 800
3 1/2	2500	8	1 1/2	R 38	960	407 200
4	2500	8	1 1/2	R 38	960	407 200

conventional sequence to assure uniform loading of the gasket. For further information on torque loading, see Section 5.2.

3.2. Flanges

All process flanges in HRE-2 are of type 347 stainless steel and of the weld-neck ring-joint type (see Fig. 7). The 1500-lb series is used in sizes 3/4 in. and below, and the 2500-lb series is used in sizes 1 in. and above. Although 1500-lb flanges are acceptable for service at 2770 psi at 600°F under the American Standard for Steel Pipe Flanges and Flanged Fittings,[1] the more conservative practice of limiting flange working pressures to their primary service pressure rating was followed.

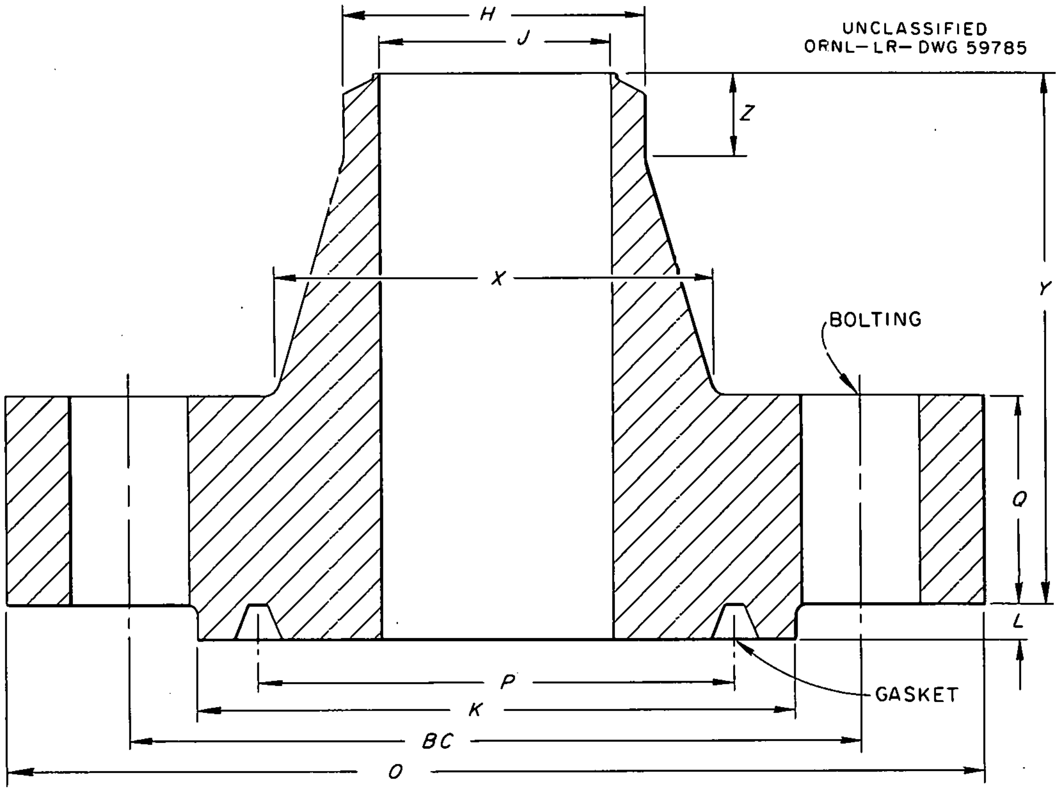
Originally, flanges and rings procured commercially to ASA tolerances (ring pitch diameter, ±0.007 in.; groove pitch diameter, ±0.005 in.) were matched as closely as possible, with the result that mismatch between the pitch diameters of ring and groove was generally within ±0.004 in. The pitch-diameter gages developed concurrently with the flange program (see sec. 4.4) permitted precise measurements to be made conveniently, both in the production and in the inspection processes. With the aid of these gages, replacement flanges for HRE-2 were produced at a later date to a pitch-diameter tolerance of ±0.001 in. (rings, ±0.002 in.) without excessive cost.

Requirements carried in current HRP specifications as a result of experience in the flange development program and in procurement are:

1. The sides of ring grooves shall have 32-μin. rms finish. (In the ASA Standard the grooves should be free of objectionable marks.)

2. Each flange shall be tested for hardness. Hardness shall be determined at a point on the face of the flange adjacent to but not on the raised face. Minimum hardness shall be Rockwell B84 or Brinell 160. (ASA flange hardness is not specified.)

3. The angle of each side of each ring groove, as measured relative to the flange face, shall be within a tolerance of ±15 min of arc. To facilitate inspection, the raised portion of the flange face is required to have a 63-μin. rms finish. (ASA requires ±30 min.)



Flange	Dimensions (in.)											Bolting		Gasket
	O	BC	K	P	X	J	H	Y	Q	L	Z	No.	Size (in.)	
1/2 in., 1500 lb	4 3/4	3 1/4	2 3/8	1 3/16	1 1/2		0.84	2 3/8	7/8	1/4		4	3/4	R-12
1 in., 2500 lb	6 1/4	4 1/4	3 1/4	2 3/8	2 1/4		1.32	3 1/2	1 3/8	1/4		4	7/8	R-18
2 in., 2500 lb	9 1/4	6 3/4	5 1/4	4	3 3/4		2.38	5	2	5/16		8	1	R-26
3 1/2 in., 2500 lb	13	8 3/4	8	6 3/16	5 7/8	3.138	4	7	2 3/4	7/16	1 1/8	8	1 1/2	R-38
4 in., 1500 lb	12 1/4	9 1/2	7 5/8	6 3/8	6 3/8		4.5	4 7/8	2 1/8	5/16		8	1 1/4	R-39
4 in., 2500 lb	14	10 3/4	8	6 3/16	6 1/2		4.5	7 1/2	3	7/16		8	1 1/2	R-38

Fig. 7. Common HRE-2 Weld-Neck Ring-Joint Flanges.

4. Average pitch diameter, as indicated by measuring three radii oriented 120 deg apart, shall be within ±0.0010 in. of the nominal pitch diameter. (ASA requires ±0.005 in.)

5. Tolerance on the radius at the bottom of the flange ring groove shall be plus 1/64 in., minus zero. (ASA does not specify a tolerance.)

3.3 Gaskets

Gaskets are of the ring-joint type (Fig. 8), both oval and octagonal cross-sectional shapes being used in the tests and in the reactor. They are fabricated of type 304 stainless steel. Some of the early gaskets were gold plated for corrosion resistance and to facilitate sealing, but the tendency of the gold plate to flake off forced abandonment of this approach. Two holes were drilled in each gasket to connect the flange groove annuli to permit the monitoring of a complete flange joint with a single leak-detector connection. Gaskets were selectively mated to the flange grooves for minimum mismatch of pitch diameters. With the pitch-diameter gages, the HRE-2 replacement gaskets were produced to a pitch-diameter tolerance of ±0.001 in. without difficulty, although the specification called for ±0.002 in.

Current specification requirements for gaskets are:

1. The material shall have a maximum hardness of Brinell 140 at a 3000-kg load (vs Brinell 160 for ASA).

2. Sealing surfaces of the gaskets shall have a 32-μin. rms finish (vs "free of defects" for ASA).

3. The average pitch diameter, as indicated by measuring three diameters 60 deg apart, shall be within ±0.002 in. of the nominal diameter (vs ±0.007 in. for ASA).

4. For octagonal gaskets the angle of seating, as measured relative to the flat face of the ring, shall be within ±0.25 deg of the nominal seating angle (vs ±0.5 deg for ASA).

3.4. Bolts

Bolts of the type used in the flange tests and originally specified for HRE-2 are shown in Fig. 9. Metallurgical difficulties described in

Sections 3.7 and 3.8 led to material and design modifications, however, and the final configuration is shown in Fig. 10. The shanks of the bolts were turned down to the thread root diameter to increase the elasticity of the systems, and the ends were pointed to facilitate remote installation. Regular bolts with extra high regular heads (rather than heavy heads) were

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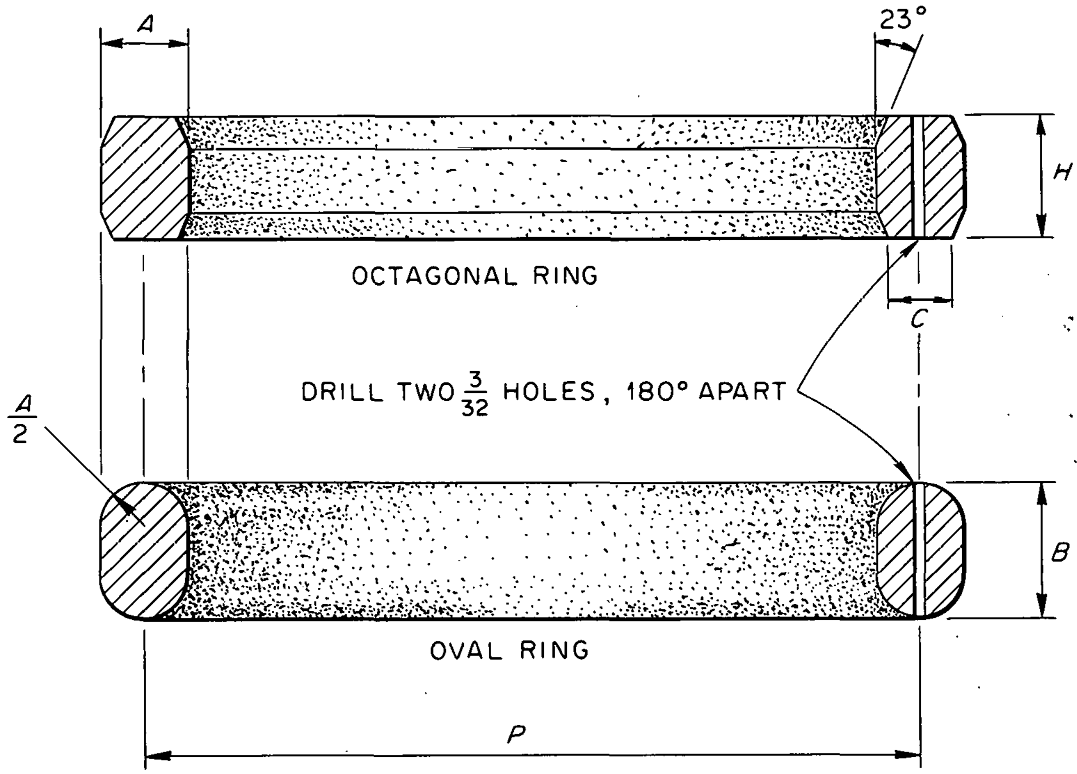

Ring No.	Dimensions (in.)				
	P	A	B	H	C
R-12	1 9/16	5/16	9/16	1/2	0.206
R-18	2 3/8	5/16	9/16	1/2	0.206
R-26	4	7/16	11/16	5/8	0.305
R-38	6 3/16	5/8	7/8	13/16	0.413
R-39	6 3/8	7/16	11/16	5/8	0.305

Fig. 8. Common HRE-2 Ring-Joint Gaskets.

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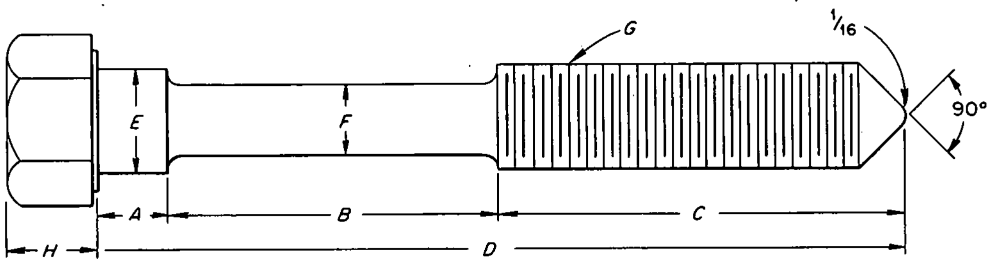

Mark No.	Bolt Size (in.)	Dimensions (in.) A	B	C	D	E	F	G	H	Flange Size	Ferrule Mark No.
1	1/2	1/4	1 1/4	1 1/2	3	1/2	0.40	1/2 13NC7	1/2	1/2 in., 300 lb	A
2	5/8	1/4	1 1/2	1 3/4	3 1/2	5/8	0.51	5/8 11NC7	5/8	3/4 in., 300 lb	B
3	5/8	1/4	1 1/2	2	3 3/4	5/8	0.51	5/8 11NC7	5/8	1 in., 300 lb	B
4	3/4	1/2	1 3/4	2 1/2	4 3/4	3/4	0.61	3/4 10NC7	3/4	1/2 in., 1500 lb	C
5	3/4	1/2	2 1/4	2 3/4	5 1/2	3/4	0.61	3/4 10NC7	3/4	1/2 in., 2500 lb	D
6	3/4	1/2	2 3/8	2 7/8	5 3/4	3/4	0.61	3/4 10NC7	3/4	3 in., 300 lb	E
7	3/4	1/2	2 1/2	3	6	3/4	0.61	3/4 10NC7	3/4	4 in., 300 lb	E
8	7/8	1/2	2 1/8	2 7/8	5 1/2	7/8	0.71	7/8 8NC7	7/8	1 in., 1500 lb	F
9	7/8	1/2	2 1/2	3	6	7/8	0.71	7/8 9NC7	7/8	1 in., 2500 lb	G
10	1	3/4	2 3/8	2 7/8	6	1	0.81	1 8NC7	1	1 1/2 in., 1500 lb	H
11	1	3/4	3 5/8	3 5/8	8	1	0.81	1 8NC7	1	2 in., 2500 lb	J
12	1 1/8	3/4	4	3 1/4	8	1 1/8	0.90	1 1/8 8N7	1 1/8		K
13	1 1/4	1	3 3/4	4 1/4	9	1 1/4	1.00	1 1/4 8N7	1 1/4	4 in., 1500 lb	L
14	1 1/4	1	4 1/2	5	10 1/2	1 1/4	1.00	1 1/4 8N7	1 1/4	3 in., 2500 lb	M
15	1 3/8	1	5 1/2	6	12 1/2	1 3/8	1.10	1 3/8 8N7	1 3/8	6 in., 1500 lb	N
16	1 1/2	1	4 7/8	5 5/8	11 1/2	1 1/2	1.20	1 1/2 8N7	1 1/2	3 1/2 in., 2500 lb	P
17	1 1/2	1	5 1/4	6	12 1/4	1 1/2	1.20	1 1/2 8N7	1 1/2	4 in., 2500 lb	Q
18	1 1/2	1	5 1/4	7	13 1/4	1 1/2	1.20	1 1/2 8N7	1 1/2	4 in., 2500 lb SP	R
19	1 1/8	3/4	4	5 1/4	10	1 1/8	0.90	1 1/8 8N7	1 1/8	2 1/2 in., 2500 lb SP	K

Tolerance: ±0.010 Decimals
±1/32 Fractions

Fig. 9. Original HRE-2 Bolts.

specified to facilitate handling with existing remote-maintenance tooling at HRE-2.

The bolts used in the HRE-2 were purchased in accordance with ASTM A193-55T, grade B7, AISI type 4140 steel, except that the steel was to be aluminum-killed, with a minimum tensile strength of 115 000 psi and a minimum yield strength of 100 000 psi. The minimum elongation in 2 in. was specified as 20%, and tempering at 1200°F with an oil quench or blowdown with air was mandatory. The target hardness was Rockwell C-24 to -30.

Special inspection requirements included the use of magnetic-particle inspection (Magnaflux) after preliminary machining and heat treatment, check tensile tests, and 100% dimensional inspection of threads, which were of

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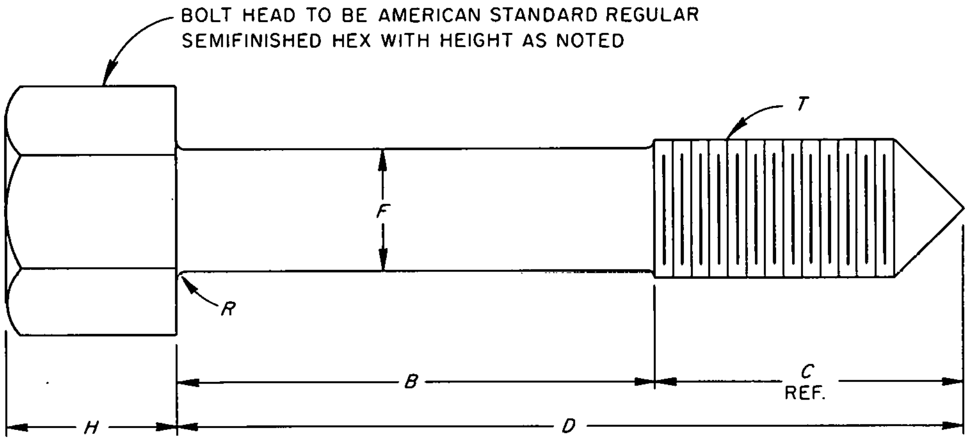

Dimensions (in.)
H B C D F R T

Mark	Service	Bolting		Dimensions (in.)						
		No.	Size (in.)	H	B	C	D	F	R	T
AA	1/2 in., 300 lb	4	1/2	1/2	1 1/2	1 3/4	3 1/4	0.40	1/16	1/2 13UNC
BB	3/4 in., 300 lb	4	5/8	5/8	2	2	4	0.51	1/16	5/8 11UNC
	1 in., 300 lb	4	5/8	5/8	2	2	4	0.51	1/16	5/8 11UNC
CC	1/2 in., 1500 lb	4	3/4	3/4	3	2 1/2	5 1/2	0.61	1/16	3/4 10UNC
	1/2 in., 2500 lb	4	3/4	3/4	3	2 1/2	5 1/2	0.61	1/16	3/4 10UNC
	3/4 in., 2500 lb	4	3/4	3/4	3	2 1/2	5 1/2	0.61	1/16	3/4 10UNC
	3 in., 300 lb	8	3/4	3/4	3	2 1/2	5 1/2	0.61	1/16	3/4 10UNC
	4 in., 300 lb	8	3/4	3/4	3	2 1/2	5 1/2	0.61	1/16	3/4 10UNC
DD	1 in., 1500 lb	4	7/8	7/8	3 1/2	2 3/4	6 1/4	0.71	1/16	7/8 9UNC
	1 in., 2500 lb	4	7/8	7/8	3 1/2	2 3/4	6 1/4	0.71	1/16	7/8 9UNC
EE	2 in., 2500 lb	8	1	1	5 3/4	2 1/2	8 1/4	0.81	1/16	1 8UNC
FF	3 in., 2500 lb	8	1 1/4	1 1/4	7 1/2	3	10 1/2	1.00	1/8	1 1/4 8N
GG	3 1/2 in., 2500 lb	8	1 1/2	1 1/2	8 1/4	3 3/4	12	1.20	1/8	1 1/2 8N
	4 in., 2500 lb	8	1 1/2	1 1/2	8 1/4	3 3/4	12	1.20	1/8	1 1/2 8N
HH	2 1/2 in., 2500 lb SP	8	1 1/8	1 1/8	8	2 3/4	10 3/4	0.90	1/8	1 1/8 8N
JJ	4 in., 2500 lb SP	8	1 1/2	1 1/2	10 3/4	3 1/2	14 1/4	1.20	1/8	1 1/2 8N
KK	4 in., 2500 lb NF	8	1 1/2	1 1/2	6	3 1/2	9 1/2	1.20	1/8	1 1/2 8N

Fig. 10. Present HRE-2 Bolts.

the unified thread form, class 2A fit, in accordance with ASA Standard B1.1-1949. Experience in procurement of these bolts subsequently led to the use of an AISI type 4340 steel for replacements to assure more uniform response to heat treatment. Experience also led to relaxation of the tensile strength and elongation requirements.

3.5. Ferrules

The ferrules were made of AISI type 4130 steel to permit the welding of the ferrules to the flange, since type 4130 steel would have a more favorable response to localized heating than type 4140 steel (the bolt material). The first ferrules had "feet" (see Fig. 11) to span the bolt hole. In practice, however, a corrugated spacer was placed inside the ferrule and around the shank of the bolt for centering. This permitted redesign to the configuration shown in Fig. 12, which also helped alleviate the metallurgical difficulties described in Section 3.7.

The physical-property, testing, and inspection requirements were similar to those for the bolts (see sec. 3.4). Subsequent experience in manufacture and use of the ferrules indicated the desirability of a change to an AISI type 4340 steel, as in the case of the bolts.

3.6. Nuts

Heavy, semifinished hexagonal nuts were purchased to ASTM A194-55T, grade 2H, with unified thread form, class 2B fit, in accordance with ASA B1.1-1949. Special inspections included magnetic-particle tests, tensile tests, and dimensional checks. The requirements for an aluminum-killed steel were relaxed. A maximum hardness of Rockwell C-30 was specified. Metallurgical and corrosion difficulties are discussed in Sections 3.7 and 3.8.

3.7. Metallurgical Difficulties

Several of the bolts ordered for HRE-2 failed in service in flange test No. 2. Subsequent observation of the bolts, nuts, and ferrules concurrently installed in the reactor disclosed that a number of those had severe cracks. An investigation of the metallurgical history of the

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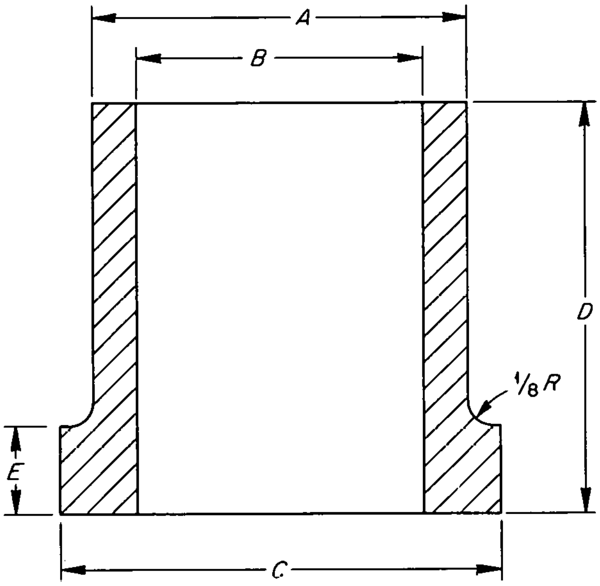

Ferrule Mark No.	Bolt Size (in.)	Dimensions (in.)					Bolt Mark No.
		A	B	C	D	E	
A	1/2	0.74	0.63	1 1/4	1/2	3/16	1
B	5/8	0.91	0.75	1 1/4	5/8	3/16	2, 3
C	3/4	1.07	0.88	1 1/2	7/8	3/16	4
D	3/4	1.07	0.88	1 1/2	1	3/16	5
E	3/4	1.07	0.83	1 1/2	1 1/8	3/16	6, 7
F	7/8	1.24	1.00	1 3/4	1	3/16	8
G	7/8	1.24	1.00	1 3/4	1 1/8	3/16	9
H	1	1.40	1.13	1 3/4	1 1/8	3/16	10
J	1	1.40	1.13	1 3/4	1 5/8	3/16	11
K	1 1/8	1.58	1.25	2	1	3/16	12
L	1 1/4	1.75	1.38	2 1/4	1 7/8	1/4	13
M	1 1/4	1.75	1.38	2 1/4	2 1/4	1/4	14
N	1 3/8	1.93	1.50	2 1/2	2 7/8	1/4	15
P	1 1/2	2.11	1.63	2 1/2	2 1/2	1/4	16
Q	1 1/2	2.11	1.63	2 1/2	2 3/4	1/4	17
R	1 1/2	2.11	1.63	2 1/2	1 5/8	1/4	18

Fig. 11. Original HRE-2 Ferrules.

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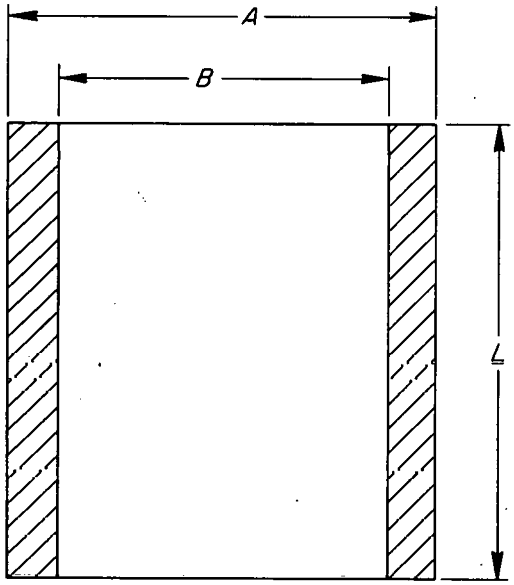

Ferrule Mark No.	Bolt Size (in.)	Dimensions (in.)			Bolt Mark No.
		A	B	L	
AA	1/2	3/4	5/8	1/2	AA
BB	5/8	15/16	3/4	5/8	BB
CC	3/4	1 1/8	7/8	1	CC
DD	7/8	1 1/4	1	1 1/8	DD
EE	1	1 7/16	1 1/8	1 5/8	EE
FF	1 1/4	1 3/4	1 3/8	2 1/4	FF
GG	1 1/2	2 1/16	1 5/8	2 1/2	GG, JJ or KK
HH	1 1/8	1 9/16	1 1/4	2	HH

Fig. 12. Present HRE-2 Ferrules.

various items revealed that the mechanism responsible for cracking was an improper quenching technique, which caused cracks at regions of stress concentration (corners).[3] As a result of this investigation, it was

[3]J. P. Hammond, G. M. Adamson, and T. M. Kegley, Cracks in HRT Flange Bolts and Ferrules, ORNL CF-57-1-163, Jan. 29, 1957.

recommended for bolts, nuts, and ferrules that:

1. aluminum-killed (deoxidized) steel be specified to assure a more uniform structure and minimize possible microfissures;
2. an oil or air-blowdown quench be specified to minimize the severity of the quenching operation;
3. metallic coatings be eliminated;
4. a minimum radius under the head and a rounded thread form be specified to eliminate stress raisers;
5. a greater than normal elongation be specified at the minimum tensile properties, implying greater care and selectivity in the heat treatment;
6. Magnaflux inspection of the bolt and nut blanks be performed after heat treatment to reject items cracked in manufacture.

3.8. Corrosion-Resistant Coatings

Because the bolts of HRE-2 are immersed in water during maintenance operations, a protective coating is desired to minimize rusting, with resultant contamination of the shielding water and possible seizure of the nuts on the bolts. Six coatings were tested by successively immersing the bolts in water, heating to about 570°F, and furnace cooling; the coatings were:

1. cadmium plate,
2. zinc plate,
3. Bonderize (iron-manganese phosphate),
4. Pentrate Super-Black (properietary black-oxide coating of Heatbath Corp.) plus a soluble oil,
5. aluminum paint,
6. molydenum disulfide (Molycoat).

The metal platings were eliminated because of the fear of deleterious metallurgical interactions. The most successful of the remaining coatings was the Pentrate Super-Black plus soluble oil, applied in accordance with the manufacturer's recommendation.

Experience with the Pentrate coated bolts in HRE-2 showed, however, that this coating was inadequate to prevent rusting over an extended period in the reactor environment. A further investigation of coatings is being made, specifically coatings of zinc (melting point, 787°F), nickel (melting point, 2651°F), and ceramic.

3.9. Insulation

Insulation specified for HRE-2 consists of a wrapping of ten layers of crinkled 0.015-in. aluminum foil with an aggregate thickness of 3/4 in.

4. Experimental Equipment

4.1. Extensometers

The problem of accurately monitoring bolt stress through thermal cycles is not simple, and several techniques were tried during the flange test program. In flange test No. 1, Baldwin high-temperature strain gages were used. The rigors of the test were too much for most of the gages, and too few survived to yield useful data. The leads were carried out singly between the flanges, and the susceptibility of the leads to damage, both during disassembly and during operation, seems to have been the cause of the failures.

In the furnace tests (see sec. 5.1) which were carried out at relatively low temperatures, it was possible to use paper strain gages, and an improved method was used (Fig. 13) for protecting the leads from mechanical damage. Here the lead wires were carried through the center of the bolt in a cable and terminated in a cable connector at the point of the bolt. This arrangement proved to be satisfactory and could be applied to the high-temperature gages to make them much more rugged.

For field use at the HRE-2 site, a C-clamp extensometer was designed and built. This device consisted of a large C-clamp with a hardened anvil at one end and a dial indicator at the other. The difficulty of maintaining a consistent surface or indentation on the ends of the bolts, of protecting the contacting anvils of the extensometer, and of obtaining consistent readings under field conditions rendered the device useless.

In flange test No. 2, pin extensometers (Fig. 14) were used. These consisted of a steel pin in a hole along the axis of the bolt that was

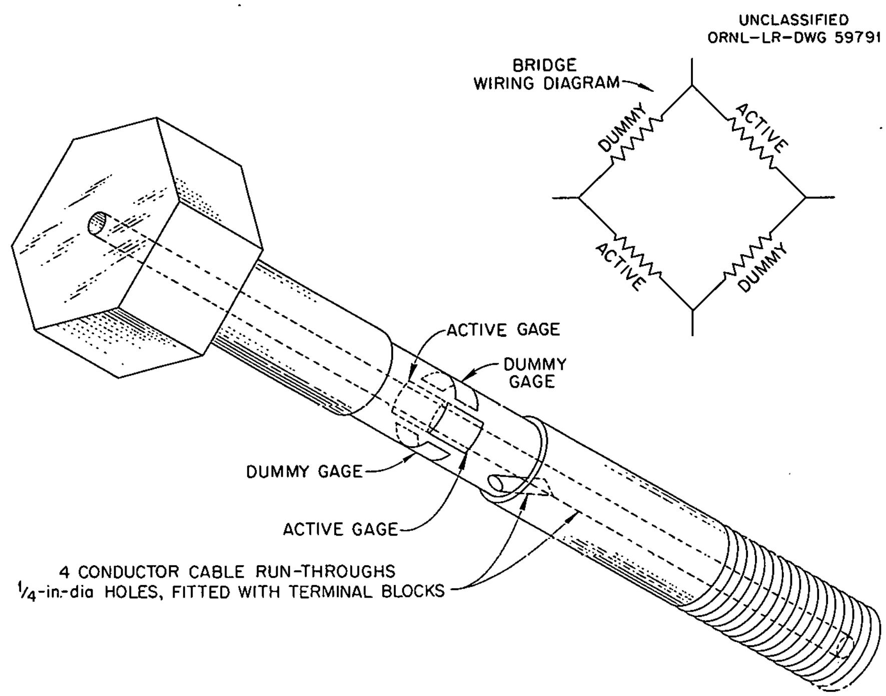

Fig. 13. Strain-Gage Bridge Used on Bolts for Easy Disconnect.

welded at the point of the bolt and projected slightly beyond the head of the bolt. Both the free end of the pin and the head of the bolt were machined normal to the bolt axis so that a dial indicator equipped with a special foot could measure the distance between the two surfaces. Since the pin was not subject to applied stresses, the differential movement of the bolt head with respect to the pin was equal to the integrated strain in the bolt, neglecting thermal expansion. Cycles 240 through 248 of flange test No. 2 were devoted to thermal calibration of the pin extensometers. During these cycles, the joint was made up and was cycled several times (to bring the bolts to their asymptotic load); then, with the assembly at pressure and temperature, one bolt was removed and its indicated change in elongation between the heated and room-temperature conditions was observed. These indicated changes in elongation and the corresponding thermal bolt load are given in Table 7. Since the pins and the bolts were of carbon steel having similar thermal expansion coefficients, the only explanations for the observed change in elongation are that the bolt and pin were not at the same temperature (a difference of

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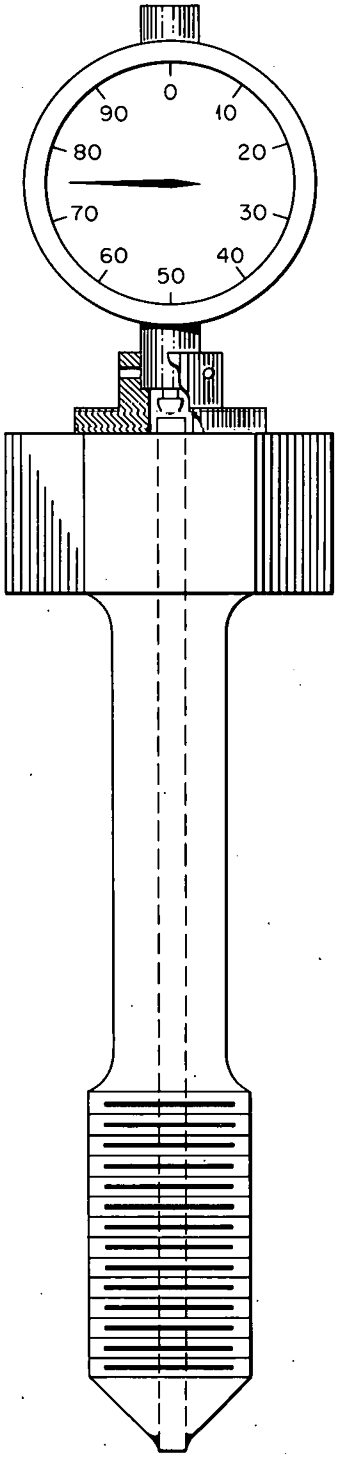


Fig. 14. Pin Extensometer — Sensor in Place on Bolt.

less than 45°F would explain away the worst case) and that heating of the sensor introduces error into the reading. It is believed that the observed change in elongation results from a combination of these two factors and could be eliminated by an improved sensor design and by allowing a longer hold time to achieve thermal equilibrium before taking pin-extensometer readings.

In general, the pin extensometers were found to be rugged and reliable, and they have performed very satisfactorily, although they are rather expensive for use in other than an experimental application. Since there was no practical extensometer available for field use, torque load tests (see sec. 5.2) were conducted to establish the reliability of torque as a measure of induced bolt load for field use.

4.2. Flange Deflection Meter

In the early stages of the program, it was postulated that excessive nonparallel deflection of the flange faces could induce leakage by reducing the contact stress of one of the two sealing circles. To investigate this, a device to measure the relative movement of two mating flanges was designed. The device consists of two spring-loaded linear potentiometers mounted on a bar fastened to one of the flanges. These potentiometers follow the relative movement of the bar on which they are mounted and a second bar mounted on the mating flange. The change in resistance produced by the movement of the slide wires of the potentiometers is measured with a Wheatstone bridge circuit.

Table 7. Apparent Thermal Calibration for Pin Extensometers

	Flange Size and Number						
	1/2 in., 1500 lb, No. 7	1 in., 2500 lb		2 in., 2500 lb		3 1/2 in., 2500 lb	
		No. 2	No. 3	No. 1	No. 4	No. 5	No. 8
Ferrule	No	Yes	No	Yes	No	No	Yes
Apparent reduction in bolt elongation in cooling from 637 to 70°F at zero bolt load, mils	0.9	2.1	1.7	2.15	2.35	2.4	2.14
Bolt load equivalent to apparent change in elongation, lb	8 800	17 400	17 000	36 800	49 500	81 700	59 200

Data from one of the devices installed on a 3 1/2-in., 2500-lb flange joint was compared with data from two 0.001-in. dial indicators during the bolting-up operation and were found to be at least as accurate as the indicators used. The device can apparently measure flange deflections to ±0.0002 in. and flange rotations to ±0.5 min.

The installation on the 3 1/2-in., 2500-lb flanged joint indicated that a bolt load of 145 000 lb (18 000 lb per bolt) induced a rotation of 4 min and a deflection (at the edge of the flange) of about 0.025 in. Since the flanges used in flange test No. 2 did not leak excessively, this line of investigation was not pursued any further.

4.3. Leak Detectors

The discussion of leak detectors presented here is restricted to the type of detection and measurement used in the flange development program and to the reduction of data recorded with this type of detection. The leak-detection technique adopted for the experimental program consisted of observing the rate of pressure rise in the ring groove annulus after evacuating and sealing off the annulus. The system is shown schematically in Fig. 15; it consists of tubing connecting the annulus to a thermocouple-type vacuum-gage tube and to a vacuum valve. Two holes are drilled through the ring gasket to connect the two annuli and to permit monitoring of the

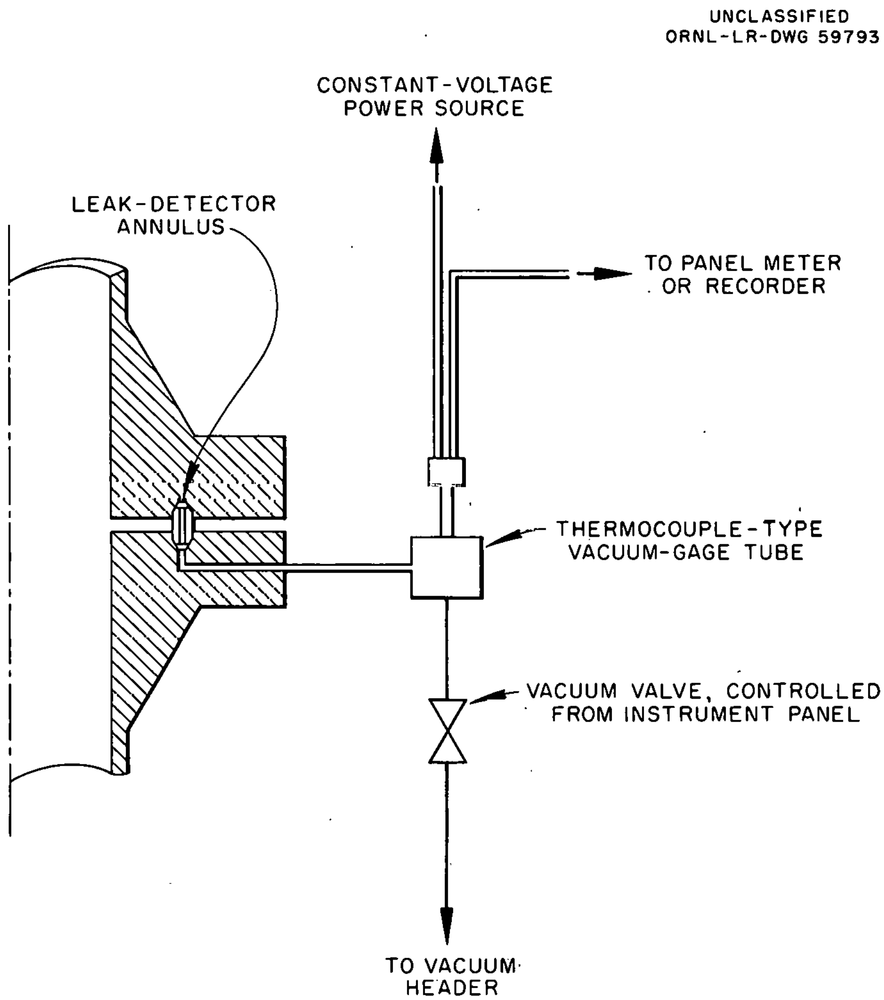


Fig. 15. Schematic Diagram of Leak-Detector System.

entire joint with a single leak detector. This technique is incapable of determining whether the indicated leakage is from the process side to the annulus or from the atmosphere to the annulus, but since satisfactory operation of the reactor leak-detector system requires a tight seal on both seating surfaces, the exact leakage path is of secondary interest.

In all the work reported, flanges were maintained in the vapor phase of the loop, and leak rates are reported in grams of water per day. The thermocouple-type vacuum-gage tubes used were calibrated under both static and transient conditions by metering air slowly into a known evacuated volume. The response of the tubes was found to be sufficiently uniform and consistent under transients and among different tubes to permit a single calibration curve to be used for all tubes, provided that the proper voltage (determined in the calibration) was applied to the heating element.

In the final leak-detector installation for flange test No. 2, the vacuum valves were equipped with air-operated controllers and were arranged so that they could be operated from the control panel. The vacuum-gage tubes were connected to a recorder through a gang switch so that individual rates of pressure rise could be recorded. For very high leak rates, the vacuum-gage tubes were switched to a panel meter, and time between two pressures was read with a stop watch.

The leak rate was calculated by using the perfect-gas law, assuming an appropriate average temperature for the evacuated volume. The following volumes were calculated for the leak-detection systems of the flange tested:

	Leak-Detection System Volume (cm^3)
1/2-in., 1500-lb flange	49
1-in., 2500-lb flange	51
2-in., 2500-lb flange	60
3 1/2-in., 2500-lb flange	90

4.4. Pitch-Diameter Gages

During the procurement of components for HRE-2 and during the early stages of the flange development program, it was observed that the techniques available for the dimensional inspection of ring-joint gaskets and flange grooves were inadequate. This inadequacy stemmed from HRE-2 requirements for a double seal, since the double seal necessitated closer tolerances than were normally used, although, in general, tools were not available to demonstrate strict conformance even to ASA tolerances.

Inquiry showed that commercially used inspection techniques included (1) gasket pitch diameter measurement with a modified vernier caliper, a slow, tedious operation with inadequate sensitivity, or fitting of the gasket to a brass master groove by touch; (2) flange-groove pitch diameter measurement by a plug gage or comparison with a male master template by use of backlighting, techniques which lack sensitivity; and (3) measurement of the bevel angles of octagonal gaskets or of the ring groove with a machinist's protractor, a satisfactory technique. It further seemed that, since complaints from users were rare and the available equipment was inconvenient to use, physical inspection of gaskets and grooves was usually of a cursory nature.

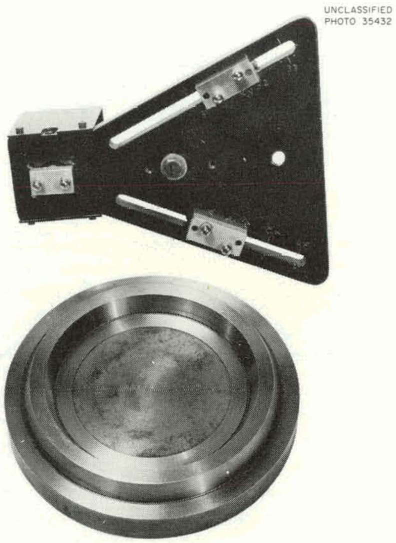
It was believed that improved inspection equipment was necessary, not only for experimental measurements of test flanges, but for inspection of HRE-2 components. Criteria were prepared and proposals were sought from several gage manufacturers, with the result that a contract was entered into with Federal Products Corporation of Providence, Rhode Island. Federal designed and built a set of pitch-diameter gages (Fig. 16) which have proved to be accurate and easy to use. The set consists of four gages and covers pitch diameters from R11 (1 11/32 in.) to R67 (18 1/2 in.) in the following ranges:

Gage No.	Range
1	R11-R20
2	R21-R37
3	R38-R47
4	R48-R67

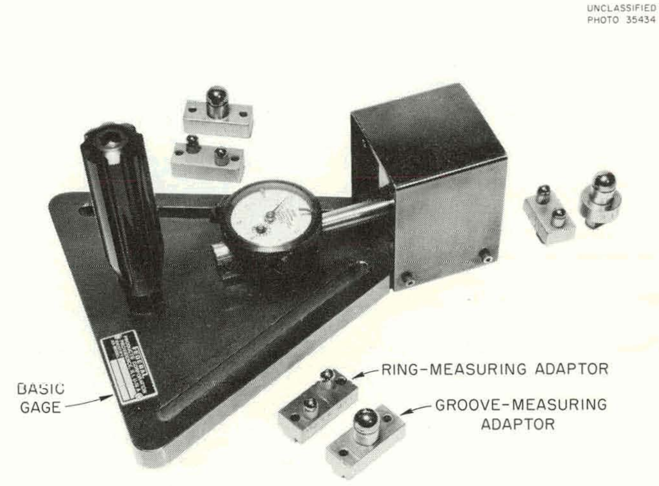
An attachment has been made to allow gage 3 to measure an R54 diameter.

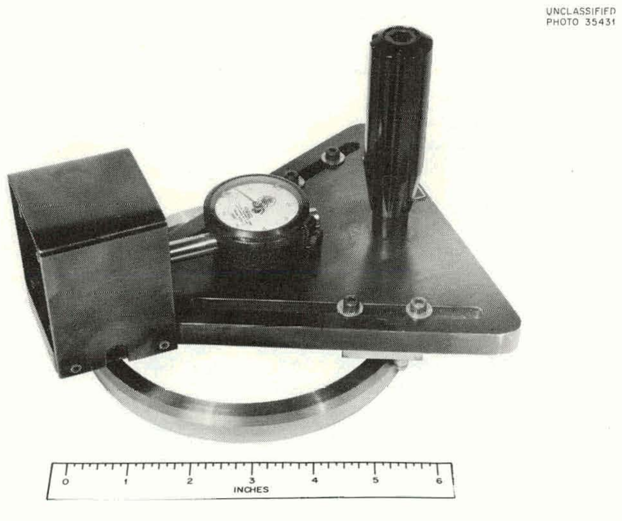
These gages are of the comparator type and are used by first attaching the contacting anvils appropriate to the surface to be gaged (male or female and to match width of gasket or groove) and then placing the gage on a master plate and setting the dial indicator to zero. The gage is then placed on the part to be measured, and the deviation of the dial indicator from zero is 1.250 times the actual deviation of the measured pitch diameter from the master. The devices indicate variations of pitch diameter as the translation of one contact point with respect to two fixed contact points. A calculated calibration indicated that the gages can accurately measure pitch diameters to ±0.0001 in. Master ring grooves and master rings have been fabricated in the sizes R11 through R40, R50, R54, and R66.

Federal also designed and built a gage to measure the bevel angle of octagonal gaskets or of flange grooves. These gages proved to be adequate but were cumbersome and difficult to use; so a machinist's protractor was used for actual gaging.



TYPICAL RING MASTER AND GAGE





TYPICAL RING GAGE IN USE

Fig. 16. Pitch-Diameter Gages.

5. Auxiliary Tests

5.1. Furnace Tests

Concurrently with the operation of flange test No. 1, tests were conducted to determine the effect of bolt shank diameter (a method of controlling resilience) on thermal loads in flanged joints. These tests were conducted in a furnace at relatively low temperatures; so the effects of internal pressure and temperature gradients were not present.

The tests were conducted on a pair of 3 1/2-in., 2500-lb stainless steel flanges with an oval stainless steel gasket and grade B7 studs. In the tests the joint was made up several times by using bolts with shanks of different nominal diameters. The bolts were instrumented with SR-4 strain gages, and the joints were cycled from room temperature to 178°F. The joint was cycled in a furnace at a rate that was sufficiently slow to keep the flanges virtually isothermal. The change of bolt load with temperature was recorded. The data from the four runs are summarized in Table 8.

Table 8. Results of Bolt Load Relaxation Tests

		Run No.			
		1	2	3	4
1.	Bolt shank diameter, in.	1.332	1.5	1.332	1.000
2.	Total bolt area (8 bolts), in.^{2}	11.1	14.15	11.1	5.28
3.	Initial load, lb	93 800	151 500	126 500	144 500
4.	Theoretical thermal load, lb	45 500	50 400	45 500	33 300
5.	Actual thermal load, lb	12 000	35 000	22 100	7 650
6.	Final load, lb	62 500	104 100	101 900	123 000
7.	Load lost during test (item 3 minus item 6), lb	31 300	47 000	24 600	21 500
8.	(Item 3 minus item 6) as % of (item 3 plus item 4)	29.6	25.2	16.6	14.1
9.	(Item 3 minus item 6) as % of (item 3 plus item 5)	22.5	23.3	14.3	12.1

The load lost during each run, as a percentage of the maximum (initial plus thermal) load, decreases with successive runs, which may be attributed to the flattening of the surface of the oval gasket and its subsequent ability to carry greater loads without plastic deformation. Although the actual thermal loads do not compare very closely with the theoretical thermal loads, they do indicate the effectiveness of increased resiliency in reducing the magnitude of thermal loads. The failure of actual loads to equal theoretical loads is attributed to the elastic-plastic action of the oval gasket and the resultant variable contact area.

Although the experiment was not carried far enough to establish the optimum area of bolt shank, it did indicate that the optimum shank area was smaller than that nominally used with the flange. It was decided to use bolts with shanks turned down to diameters approximately equal to the root thread diameter of the bolt and to increase the resiliency of the bolting further by the addition of a ferrule (a cylindrical metal collar of essentially the same area as the bolt shank) between the bolt and the flange, which would increase the effective length of the bolt and therefore its resiliency.

5.2. Torque-Load Tests

Since tightening with a torque wrench is the simplest method of controlling the tensioning of bolts in the field, attempts to establish reliable torque-load relationships were made through all the experimental work. Data from a literature survey[4-8] are presented in Fig. 17 as a plot of pounds of bolt load per foot-pound of torque for various nominal bolt diameters. In general, these data indicate that a linear relationship

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[4]P. A. Sturtevant Company, Torque Manual, pp. 23–25 (1951).

[5]Crane Company, How to Assemble and Maintain Flanged Joints, Piping Data Sheet No. 6, Valve World, No. 1, p. 16 (1952).

[6]Alex Brunner, Steel Bolts .... Determining Permissible External Axial Loads, Machine Design, pp. 124–7 (June 1950).

[7]W. C. Stewart, Determining Bolt Tension from Torque Applied to the Nut, Machine Design, pp. 209–11 (November 1955).

[8]Raymond Ei, Nomogram for Determining Tightening Torque on Bolts, Design News, pp. 39–40 (January 15, 1955).

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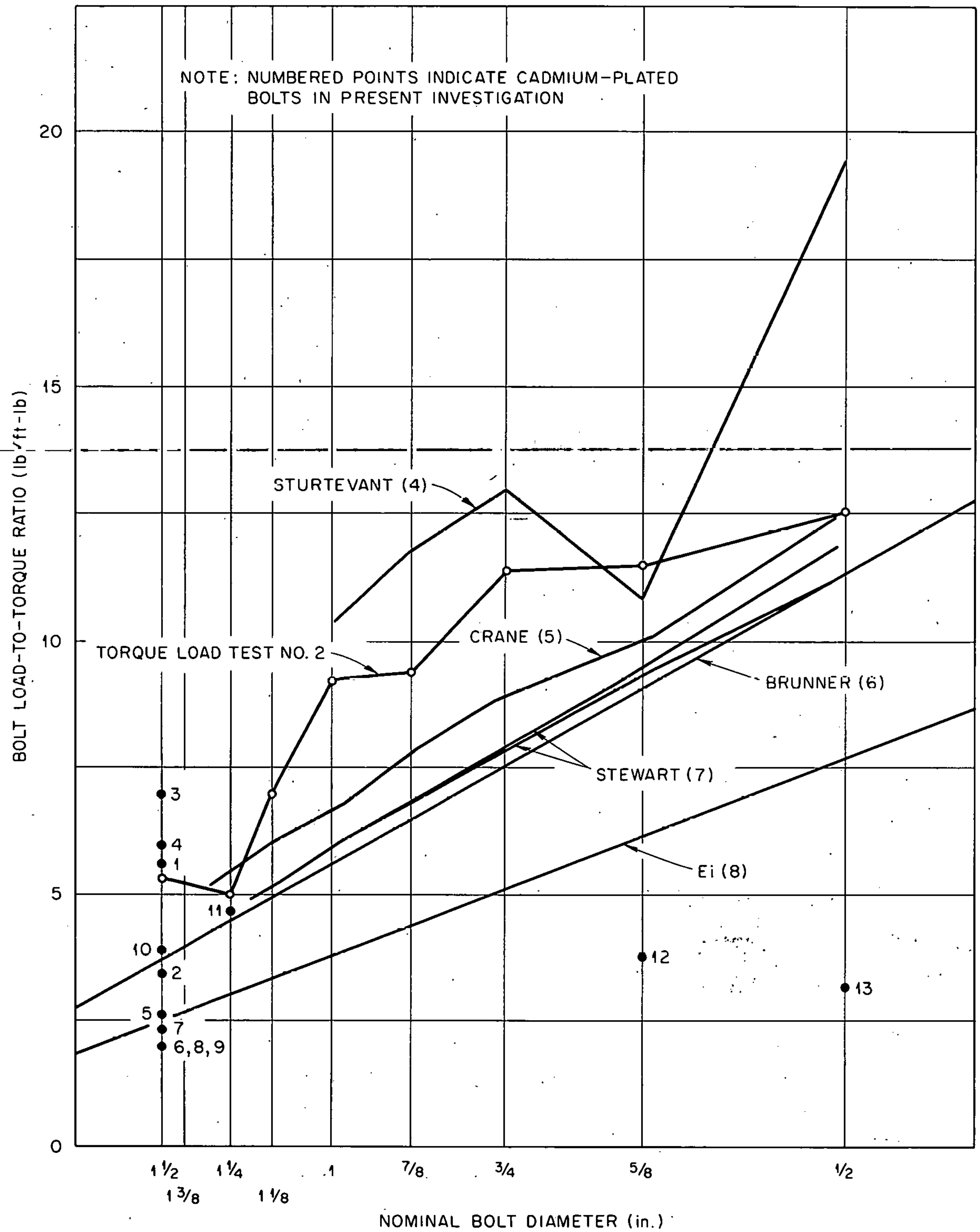

Fig. 17. Bolt Load-to-Torque Ratio vs Bolt Diameter.

may be expected between the torque-load ratio and the reciprocal of the nominal bolt diameter. Experimental data on unplated bolts compared well with the published data, but experimental data on the cadmium-plated bolts originally specified for HRE-2 (isolated data points in Fig. 17) exhibited too high a torque-to-load ratio for large bolts and too low a ratio for small bolts. This deviation is attributed to the infringement of the plating on the clearance allowed by the thread tolerances, which is more serious in the smaller sizes for a class 7 fit. In the experimental work galling and crumbling of the cadmium was observed, which would support this conclusion. In the larger sizes the plating would cause no interference with a class 7 fit, and the smoother mating surfaces would permit the high loads observed. The substitution of the new class 2A fit for the obsolescent class 7 fit relieves this condition, for the clearance in class 2A does not increase as rapidly with bolt diameter as it does in class 7.

Although the torque-to-load ratios for plated bolts did not compare closely with the available data for unplated bolts, it was observed that data for bolts of the same size and from the same shipment grouped closely. A general conclusion drawn from the tests is the bolts of a given lot will exhibit similar torque-load relationships, but small variations between lots prevent any consistent correlation from being drawn. It is therefore necessary to conduct tests on each lot of bolts when close control of bolt load is desired by using torque.

In making up the flange joints in flange test No. 2 (see sec. 2.4), incremental torque and elongation data were recorded for all the bolts. Subsequent analysis of these data indicated that torque was adequate to control bolt stress to ±10%. This was true if the condition of the bolts was consistent (i.e., if the bolts and nuts were cleaned and adequately lubricated).

It was noted during this period that the torque required to produce a given bolt stress decreased as the number of tightening-loosening cycles increased. This phenomenon is attributed to a "conditioning" of the mating surfaces of the bolt and nut, resulting from the smoothing of minute irregularities.

When the final HRE-2 bolts (with Pentrate Super-Black finish) were received, it was decided to "condition" them in the hope that this would make better field control of bolt loads possible. Each bolt-nut assembly was tightened with a torque wrench in a load cell a number of times until a constant loading was induced several successive times by the same torque. At this point the bolt was calibrated for the specified torque to be applied upon installation in the HRE-2 flanges. It is believed that, allowing for the error introduced in field bolting operations, torque wrenches can be used to tighten the conditioned nuts and bolts to loads within ±15% of the values specified. Torques determined during these tests to induce a stress of 45 000 psi (the highest stress considered reasonable) in the shank of the conditioned HRE-2 bolts are shown in Table 9. The tabulated load-to-torque ratios are plotted in Fig. 17 as a function of bolt diameter.

Table 9. Torque to Induce 45 000 psi in HRE-2 Bolts

Bolt	Torque (ft-lb)	Ratio of Load To Torque (lb/ft-lb)
1/2 in., 13UNC	45	126
5/8 in., 11UNC	80	115
3/4 in., 10UNC	115	114
7/8 in., 9UNC	190	93.8
1 in., 8UNC	250	92.8
1 1/8 in., 8N	400	71.6
1 1/4 in., 8N	700	50.5
1 1/2 in., 8N	960	53.0

5.3. Chemical Technology Division High-Pressure Flange Tests

During the period covered by the flange development program in the Reactor Experimental Engineering Division (REED), a similar effort was carried out by the Unit Operations Section of the Chemical Technology Division (CTD). Although the two efforts were conducted independently, duplication was avoided by limiting the CTD work to the small flanges with which they were concerned and the REED work described in this report to the larger flanges. Details of the CTD work will not be presented here, but the scope of the work and the conclusions reached are presented for comparison with the REED work.

The CTD work was performed on 1/2-in., 2500-lb weld-neck ring joint flanges, the flange used most commonly in the HRE-2 chemical processing plant. General areas investigated were (1) dimensional tolerances of ring gasket and flange groove, (2) surface-finish requirements for gasket-groove interface, (3) increase of bolt resiliency by the use of ferrules and by the use of Belleville washers, and (4) a comparison of oval and octagonal gaskets in the light of other considerations. With respect to dimensional tolerances and surface finish the CTD work confirmed the work reported by the REED group.

The CTD work established the necessity for introducing resiliency into the joint, and ferrules were used successfully to accomplish this. Since ferrules can make a flanged joint somewhat cumbersome, some effort was devoted to the use of Belleville washers as a compact means of introducing additional resiliency. The performance of the washers did not come up to expectations, but it was concluded that, with further development, a satisfactory washer system could be designed. The cost of such a system would have to be balanced against the expected savings in space. (Note: Although the Belleville washers may be economically feasible for small flanges, space requirements render them impractical for larger flanges; so no effort was devoted to their use in the REED program.)

In general, flanges made up with octagonal ring gaskets were more likely to be leaktight and were less likely to experience deformation of the ring groove than were those made up with oval ring gaskets. Based on this experience, it was the conclusion of the CTD group that octagonal ring gaskets should be specified for reactor chemical plant service.

Acknowledgements

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Division (now the Reactor Division). The work was under the general supervision of I. Spiewak, W. R. Gall, and M. I. Lundin and the direct supervision of I. K. Namba, P. N. Harley, R. H. Chapman, R. D. Cheverton, J. S. Culver, H. C. Roller, and J. N. Robinson.

Work performed by the Unit Operations Section of the Chemical Technology Division was under the supervision of C. C. Haws.

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