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GUIDELINES TO ACHIEVE SEALS WITH MINIMAL LEAK
RATES FOR HWR-NPR COOLANT SYSTEM COMPONENTS

by

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ABSTRACT

Seal design practices that are acceptable in pressurized-water and boiling-water reactors in the United States are not usable for the Heavy Water Reactor-New Production Reactor (HWR-NPR) because of the stringent requirement on tritium control for the atmosphere within its containment building. To maintain an atmosphere in which workers do not need protective equipment, the components of the coolant system must have a cumulative leak rate less than 0.00026 L/s.

Existing technology for seal systems was reviewed with regard to flange, elastomer, valve, and pump design. A technology data base for the designers of the HWR-NPR coolant system was derived from operating experience and seal development work on reactors in the United States, Canada, and Europe. This data base was then used to generate guidelines for the design of seals and/or joints for the HWR-NPR coolant system. Also discussed are needed additional research and development, as well as the necessary component qualification tests for an effective quality control program.

1. INTRODUCTION

The objective of this effort was to develop guidelines for designing seals with minimal leak rates for the coolant system components in the Heavy Water Reactor-New Production Reactor (HWR-NPR). Seal practices which are acceptable in pressurized-water and boiling-water reactors in the United States are not acceptable for the HWR-NPR. The nuclear power industry accepts water leak rates of the order of 0.0065 L/s from unknown sources and up to 0.065 L/s from known sources to their containment building. An upper limit is 0.65 L/s, the maximum that the recirculation pumps can handle. With a leak rate of 0.065 L/s being acceptable, the relative humidity within the containment buildings of commercial reactors is often 100%.

The Heavy Water Reactor-New Production Reactor has stringent requirements on tritium control. At a water leak rate of 0.0065 L/s (ten times more stringent than the commercial power industry), the atmospheric tritium concentration within containment can rise to 40 $\mu\text{Ci}/\text{m}^3$ in 10 min in a volume of 10^5 m^3 , at a heavy water tritium concentration of 1 Ci/L. This room concentration exists even if high-efficiency driers are in operation since economics would limit the maximum flow rate to the driers to <10 vol%/min for volumes of 10^4 to 10^5 m^3 . For an

unprotected worker, this tritium concentration results in a dose rate of 5 mrem/h, or a dose of 0.5 rem in 100 h. This dose is the annual goal for each worker at the HWR-NPR. Since the dose from gamma radiation has not been included, this is a lower limit on the dose that the worker would receive under these conditions.

The leak rate to the containment building which is acceptable for the coolant components of the HWR-NPR must meet three requirements: (1) the tritium concentration within containment is acceptable for workers, (2) the yearly heavy water losses from containment are minimized, and (3) reasonably sized air cleanup systems and deuterated water upgrade systems are adequate to achieve the first two requirements.

Regulatory requirements and/or maintenance requirements have driven the development of new seal materials and/or concepts within the United States. Environmental Protection Agency (EPA) requirements have resulted in the replacement of asbestos valve packings and gaskets with expanded graphite materials. For retrofits with expanded graphite materials, the designs of flanges and valves are not optimized since the properties of expanded graphite are significantly different from asbestos. Thus, flange or valve hardware built to operate with asbestos cannot achieve the optimum leak rate possible with expanded graphite. A new valve and/or joint design is therefore needed to achieve the lower leak rate that is possible.

In the United States published information on seal modifications which result in low leak rates is often not incorporated into specifications and/or designs for new systems because of a lack of knowledge, conservativeness in specifying new product lines, or a perceived capital cost disadvantage. Failure to incorporate existing technology (American, European, Canadian) into design practices for seal components results in high operating costs in the commercial nuclear power industry, both for replacement electrical power during repairs and for the extra worker time needed to minimize individual dose during repairs.

For the HWR-NPR, the specification of seal systems which have cumulative leak rates of 0.00026 L/s (30 times lower than any commercial specification), would be highly desirable. To specify these seal systems, existing technology was reviewed in four categories: (1) flange design, (2) elastomer design, (3) valve design, and (4) pump design. The collected data base was derived from operating experience and seal development work for commercial reactors in the United States, Canada, and Europe. Groups which have sponsored or generated sustained programs are the Electric Power Research Institute (EPRI), Atomic Energy of Canada, Limited (AECL), and the British Hydromechanical Research Association (BHRA) in Europe. Major changes in the specifications for joints or seals have arisen from their work and should be incorporated into the specifications for the coolant components of the HWR-NPR.

Our work can serve as a basis for guidelines to design a coolant system which can achieve the leak-rate requirement of the HWR-NPR design. Areas needing research and development are summarized. Qualification and verification tests for components are specified. Capital costs which might be incurred by following the recommendations of this study would be offset by the lower operating costs incurred over 40 years of HWR-NPR operation.

2. EXECUTIVE SUMMARY

The state of the art in seal development was determined by reviewing the relevant literature for the design of flanges, elastomer seals, valves, and pumps. The collected data base includes data obtained from operating experience and seal development efforts for commercial reactors in the United States, Canada, and Europe. Guidelines to direct the design of effective seals for the coolant system in the HWR-NPR were derived from this information. These guidelines are especially important in designing the flange seal on the reactor head; the dynamic seals for the fuel tubes, control rods, and safety rods which pass through the reactor head; the isolation valves; and the reactor pumps. The guidelines are divided into the following categories: engineering design, manufacturing requirements, assembly requirements, maintenance requirements, material tests, and quality control tests.

Engineering Design

Common practices for the design of bolted flanges, elastomer seals, valves, and pumps should be reassessed.

The ASME formulas used to compute the sealing load for a flange are not justified, as shown by work sponsored by the BHRA. An alternative system should be used to specify compression loads for flange seals using a spiral-wound gasket with an expanded-graphite filler.

For large diameter seals in critical locations, a finite element analysis is required to determine the rotation in the joint, which is a function of the internal pressure and the structural stiffness of the flange. The seal should be modeled full size, since deformation characteristics and contact stress are influenced by a seal's cross section.

The valve designer has several choices for the HWR-NPR plant. First, the number of valves can be minimized within the constraints of system maintenance. Second, bellows valves or welds can be used if fatigue is not a problem or if loss of access is allowed. Third, valves using die-formed expanded-graphite packing with live-loading can be incorporated. For the latter, packing and spring combinations have to be tested so that spring stacks can be designed to assure a sustained high gland load on the packing. This option probably will require the design of new valves.

For centrifugal pumps, ethylene propylene seals with a curved-back U-cup design should be used in shaft seals to minimize the effects of power blackouts on the degradation of the shaft seal.

Manufacturing Requirements

Two factors are important in manufacturing: the consistent availability of a qualified component and the consistent production of that component to specification. Certificates of conformance or compliance would be advisable to ensure that the required material is supplied. These certificates would include the manufacturing history, especially important for bolts, and would document the exclusion of deleterious materials (chloride or sulfur containing).

The tolerances needed on a flange or valve body to achieve a given leak rate must be specified. The effects of post-machining stress relaxation or the influence of other structural components need to be considered in the flange design to achieve a seal with a low leak rate. Surface waviness and changes in tolerances increase with physical size, and distortions can occur due to the weight-carrying capacity of a component like a flange.

Assembly Requirements

Special tools may be required for assembly or disassembly of flanges, valves, and/or pumps. These tools need to be specified and designed if so required. For example, an ultrasonic extensometer should be specified for the assembly of bolted flanges.

Maintenance Requirements

Achieving seals that are reliable and maintainable requires considerable effort, including the collection of failure rate data for each component and the determination of failure modes, corrective maintenance times, and preventive maintenance times. Design decisions based on analysis of failure modes can improve reliability and maintainability. Physical changes in equipment, systems, or procedures can improve reliability, as can clearly stated specifications for the equipment and factual evidence that the specifications are met.

Maintenance requirements should include not only physical maintenance but also the specification of maintenance records on each component throughout its lifetime.

Material Tests

A rigorous characterization of the properties of spiral-wound graphite-filled gaskets, ethylene propylene, and expanded-graphite packing material is needed since these properties serve as a basis for designer's calculations.

For spiral-wound gaskets, the leak behavior has to be measured using the technique developed under the sponsorship of BHRA. The properties of the elastomer (EPDM) have to be well characterized. The response to load of graphite packing and Belleville springs, as a function of time, needs to be measured so that the lifetime of these components can be estimated.

Quality Control Tests

A quality control program is needed for the design, manufacture, installation, and assembly of all coolant component seals and/or joints. Specifications must be set for the components to ensure that they achieve the minimal leak rate. Then the components have to be qualified in defined tests to ensure long-term leakage performance. In addition, procedures for the installation and assembly of these components within containment have to be documented. The four types of quality control tests necessary for all components are qualification, validation, compliance, and development tests.

3. CRUCIAL SEALS FOR HWR-NPR

The HWR-NPR coolant systems will require several crucial seals: a flange for the reactor head, static elastomer seals for the fuel rods, dynamic elastomer seals for the control and safety rods, isolation valves, and reactor pumps.

3.1 Flange for Reactor Head

The design of the flange seal on the reactor head is a major design challenge. To achieve a design which has both the strength and the flexibility needed, the best experience on flanged joint design and assembly has to be incorporated into the HWR-NPR design effort. Each feature of this seal (the design and manufacture of the flange faces, the type of gasket, the type of bolt, the method of assembly) influences the effectiveness of the seal. Therefore, we have examined each of these areas to provide for the designer information pertinent to the goal of an effective seal.

3.2 Elastomer Seals for Control Rods/Safety Rods/Fuel Tubes

The current control rods for Savannah River Site reactors are in open standpipes. Tritiated water vaporizes from these standpipes to the reactor building. For the HWR-NPR, the control rods will have a sealed drive system, and multiple rods will be interlinked.

Since these control rods have to constantly change position within the reactor core, a flexible but leak-tight seal is required at the reactor and air interface. The design of this seal should incorporate current data on elastomers which have the greatest resistance to tritium embrittlement or the loss of their compressibility. In addition, live-loaded seals, which provide a constant compression force, may be an effective way to improve the control rod's seal.

The HWR-NPR design of the fuel rod has as its greatest challenge, the refueling operation. For pressurized-water reactors (PWRs) refueling is done under water,^{1,2} but there are hardware malfunctions, many of which cause leaks. Improvements in hardware have been recommended for both PWRs³ and boiling-water reactors (BWRs)⁴ to address the fueling operations.

3.3 Isolation Valves

The HWR-NPR design has attempted to minimize the problem of leaking valves by eliminating most valves within the containment building. However, there is one set of valves which cannot be eliminated, the isolation valves. These valves have a crucial function; therefore, their design requires the incorporation of the best input from existing data. These data include specifications for live-loading and expanded graphite packing. These data have been summarized for the designer's benefit.

3.4 Pumps

Two types of pumps have been proposed for the main coolant system, centrifugal pumps and canned motor pumps. Available information for each of these systems has been reviewed. For centrifugal pumps, information relevant to the seals on the shaft has been examined, especially the behavior of this system during off-normal conditions. Recommendations are made on desirable elastomer seals to use on the pump shaft. For canned motor pumps, in which blockage of the coolant channels is a problem, the needed information for corrosion behavior of the internal graphitar bearing is mentioned.

4. SEAL REQUIREMENTS IN HWR-NPR

Specifications for seals have to conform to the low leak rate requirements for the HWR-NPR. At an early stage in the development of the design, the ability to meet these specifications must be assured.

The leak rate to containment which is acceptable for the coolant components of the HWR-NPR must meet three requirements: (1) the yearly heavy water losses from containment are minimized, (2) the tritium concentration within containment is acceptable for workers, and (3) reasonably sized air cleanup systems and deuterated water upgrade systems can achieve the first two requirements.

The influence of various parameters on seal leakage and permeation must be considered. The parameters include system pressure, system temperature, fluid properties, seal materials, seal dimensions, geometric shape of seal, surface condition, and the effects of aging.

4.1 Leak Rate

Leakage (Q_p) is a function of the ratio of the contact pressure (P_c) to material yield shear stress (k) or to material elastic modulus (E), for plastic or elastic deflections. An increase in the contact pressure results in a reduction in leakage:

$$Q_p = f(P_c/2k) \text{ or } Q_p = f(P_c/E) \quad (1)$$

where f indicates a functional relationship.

Water losses from a seal are due to permeability or defects in the seal. For elastomers, the lower leak rate limit is the permeation rate. Several workers have measured permeability for elastomers.⁵⁻⁸ The upper limit for water losses for elastomer, graphite, or metal seals is due to defects in the seal material. Defects in seal materials can increase over time due to corrosion or wear effects.

Several calculational techniques based on fluid flow theory have been developed for estimating component leakage.⁹⁻¹⁵ One technique is based on characterizing coolant leakage from a component by means of the helium leak rate for that component.⁹ A common difficulty for this technique is the selection of parameters to characterize the components in terms of the

leak rate. The most direct technique assumes that the helium leak rate is known, based on quality assurance acceptance standards or leak rate measurements at a known temperature and pressure. Component leakage can then be characterized by determining the dimensions of the leak path which would produce such a leak rate under the given conditions of temperature and pressure.

The determination of the leak-path dimensions is determined as a function of molecular or laminar flow. For laminar flow, the Poiseuille equation is used:

$$Q_l = \pi (D/2)^4 (P_a/8\mu L) (P_1 - P_2) \quad (2)$$

where Q_l is the volumetric leak rate, P_1 is the upstream pressure, P_2 is the downstream pressure, P_a is the average of the two pressures, D is the leak path diameter, L is the leak path length, and μ is the fluid viscosity. For molecular flow, the Knudsen equation is used:

$$Q_m = (2 \pi RT/M)^{0.5} (D^3/6L) (P_1 - P_2) \quad (3)$$

where R is the gas law constant, T is the absolute temperature, and M is the molecular weight of the fluid. If liquid laminar flow is assumed, the leak-path dimensions are used to calculate water leakage:

$$Q_L = \pi (D/2)^4 (P_1 - P_2)/8\mu L \quad (4)$$

To apply this technique, it is necessary to specify an appropriate helium leak rate. A leak-tightness standard, equivalent to a helium leak rate of $10^{-6} \text{ cm}^3 \text{ (STP)/s}$, is proposed for each manufactured component at its beginning-of-life for the HWR-NPR since components with this leak rate would meet the water leak rate requirements for the containment building. This leak-tightness standard can be used to estimate the chronic leakage at the beginning-of-life from components of the water coolant systems. Because these components will have different leak rates, some of which may represent a significant increase above the leak-tightness standard during operation, we assumed that 10% of the population have a leak rate two orders of magnitude higher than the standard, e.g., $10^{-4} \text{ cm}^3 \text{ (STP)/s}$.

Amesz⁹ derived the following equation for water leakage:

$$Q_{H_2O} = \frac{9.5 Q_{He} \mu_{He} (\text{density}_{H_2O}) (P_{H_2O} - 1)}{\mu_{H_2O} P_{He} (P_{He} + 31(\mu_{He}/D)) (T_{He}/M_{He})^{0.5}} \quad (5)$$

where the leakage is in g/h, the viscosity in cP, density in g/cm^3 , pressure in atm, leak diameter in μm , and helium leakage in "lusec" ($1.315 \times 10^{-3} \text{ atm cm}^3/\text{s}$).

We calculated the component leak rates using Eq. 5 and assumed a helium leak rate of $10^{-6} \text{ cm}^3 \text{ (STP)/s}$ and a water pressure of 0.5 MPa to account for the effect of transients. The results of these calculations for the effect of pressure, temperature, and leak diameter on leak rate are shown in Fig. 1. For the case of 100°C and 0.5 MPa, the water leak rate is listed in Table 1 as a function of leak diameter assumed. Also listed in Table 1 is the total tritium release if the

Component Water Leak Rate Equivalent to Helium Leak Rate of $10^{-6} \text{ cm}^3/\text{s}$

Fig. 1. Effect of Pressure, Temperature, and Leak Diameter on Water Leak Rate Calculated from Helium Leak Rate of $10^{-6} \text{ cm}^3(\text{STP})/\text{s}$

tritium concentration in the coolant water is 1 Ci/L. The last column in Table 1 is the room tritium concentration for the optimum case, since some percentage of the components during operation will have a leak rate greater than 10^{-6} cm³/s. An increase in the equivalent helium leak rate (ordinate of Fig. 1) by a factor of ten would increase the leak rate by a factor of ten. Thus, for a helium leak rate of 10^{-4} cm³/s at 0.5 MPa and 100°C, the equivalent water leak rate for a 10 μ m diameter hole is 200 mg/h for one component. If 40 (10%) of the components have this leak rate, the room tritium concentration is 19.2 μ Ci/m³. The benefit of air processing to remove tritium from the atmosphere has been ignored for this example.

Table 1. Effect of Leak Diameter on a Component's Water Leak Rate^a at 0.5 MPa and 100°C (Optimum Case)

Diameter, μ m	H ₂ O Leak Rate, mg/h	Tritium Release Rate, ^b μ Ci/d	Tritium Concentration, ^c μ Ci/m ³
0.01	0.005	48	0.0048
0.1	0.05	480	0.048
1	0.3	2880	0.288
10	2	19200	1.92
100	3	28800	2.88

^aEquivalent to a helium leak rate of 10^{-6} cm³(STP)/s.

^bConservative estimate since there will be >400 fuel assemblies, control rods, and safety rods.

^cVolume assumed to be 10⁴ m³.

Although seals and joints at the beginning-of-life can have leak rates which would not require extensive air processing within containment, the leak rate during operation is the more important parameter. To minimize the leak rate during the 30-40 years of operation of HWR-NPR, one must first optimize the design of each seal, and then ensure that the assembly, operation and the maintenance of each seal is done in accordance with qualified procedures.

The seals also have to survive conditions during an accident or a transient. (A "transient" is an off-normal event which may not require shutdown of the reactor. An "accident" is an off-normal event which requires reactor shutdown and major recovery efforts.) For example, a fuel/target tube could melt during a transient, which would increase the tritium content in the coolant. In an accident, major amounts of D₂ and T₂ could be released because of radiolysis of the coolant, metal/water interactions, or T₂ releases from target tubes. To protect from catastrophic failures, backup systems to minimize leakage may be provided on equipment. These backups include welds on bolted flange joints and on valve bodies and water collection systems for centrifugal pumps. The presence of these backups not only adds complexity to the coolant components, but also complicates their maintenance, reduces access within containment, and increases space requirements within containment.

4.2 Aging Effects

The effects of aging are difficult to assess since they can be due to material degradation, creep, and/or stress relaxation. The only study to address this issue was a two-year

effort¹⁶ which examined the relative impact that aging has on component failures in light-water reactors. (The source material in this study was the records in the coding structure of the Nuclear Plant Aging Research Program sponsored by the U.S. NRC.) The records were classified into five generic failure categories: (1) aging, (2) design and installation error, (3) testing and maintenance activities, (4) human-related actions, and (5) other (unknown or unclassified) failures. The study had to contend with several problems: (1) service ages were uncertain, (2) plant-specific effects were masked, and (3) 50% of the data records indicated unknown failures, which were classified as "other" since the causes were never determined.

Aging was found to cause 32% of all failures. The systems and components affected by aging included the pumps in component cooling systems and the valves in the containment isolation system. Also, normally operating fluid systems, like the component cooling water, tended to exhibit slightly higher proportions of aging-related failures than stand-by systems.

The causes of the failures for different reactor systems are summarized in Table 2. In Table 3, the consequences of the failures in different systems are summarized as a loss of system function, degraded operation, loss of redundancy, loss of a subsystem, and none. Loss of function accounted for 2% of the service interruptions for any reactor system.

Table 2. Causes of Failures in Reactor Systems

System	Failures, No.	Percent				
		Design	Aging	Maint.	Human	Other
Auxiliary Feedwater	829	10	30	10	1	49
Component Cooling H ₂ O	976	5	48	8	1	38
Containment Fan	318	7	26	10	1	56
Containment Isolation	926	3	45	5	1	45
Main Feedwater	827	10	38	5	1	45
Reactor Bldg. Cooling	65	8	28	10	8	46
Reactor Coolant	1012	15	28	5	1	52
Residual Heat Removal	1968	12	26	8	1	55
Service H ₂ O	1272	8	45	8	1	40

Table 3. Type of Service Interruption Caused by System Failure

Reactor System	Percent				
	Lose Function	Degrade	Lose Redun.	Lose Subsys.	None
Auxiliary Feedwater	1	19	18	32	32
Component Cooling H ₂ O	1	15	13	35	38
Containment Fan	2	15	25	25	35
Containment Isolation	0	25	15	30	34
Main Feedwater	2	20	5	18	52
Reactor Bldg. Cooling	0	20	12	32	25
Reactor Coolant	2	23	13	23	40
Residual Heat Removal	1	18	15	24	45
Service H ₂ O	0	17	19	35	30

Tables 4 and 5 present data on valve and pump failures. These tables show the percent of component failures due to aging, and the percent that these component-aging failures are represented in reactor system failures. A valve is considered failed when it does not meet any of the performance requirements in its technical specification. In the study, the confidence intervals for the valve data are small (20%), which indicates that aging in valves is independent of the reactor system in which they reside. This indicates that there is a need to examine valve design.

Table 4. Cause of Reactor System and Valve Failure Due to Aging

Reactor Systems	Valve Failure due to Age, %	Total Valve Failures, No.	System Failure due to Valve Age, %
Auxiliary Feedwater	46-58	267	17
Component Cooling H ₂ O	51-62	276	16
Containment Fan	13-35	65	5
Containment Isolation	52-63	451	28
Main Feedwater	47-58	334	22
Reactor Bldg. Cooling	28-89	10	10
Reactor Coolant	42-60	118	7
Residual Heat Removal	46-55	484	12
Service H ₂ O	51-60	362	16

Table 5. Cause of System and Pump Failure Due to Aging

Reactor Systems	Pump Failure due to Age, %	Total Pump Failures, No.	System Failure due to Pump Age, %
Auxiliary Feedwater	28	110	4
Component Cooling H ₂ O	75	183	14
Containment Fan	68	3	1
Main Feedwater	48	49	4
Reactor Coolant	35	93	4
Residual Heat Removal	48	86	3
Service H ₂ O	58	248	12

5. SEALS IN FLANGES

Seal development for flanges (bolted joints) does not require the development of a special material but the integration of design and assembly factors to ensure that sufficient clamping force is exerted to form an acceptable seal. If any factor is ignored, the seal may have a high leak rate in a short operating time. Some of the factors to be considered in seal development are material property data for gaskets, bolts, and flanges; engineering design data for flanges, which are used to calculate the clamping force required for a given flange stiffness; quality control during the manufacture of joint components, especially the finish on the flange surface; and quality control during the assembly of the joint to achieve the desired clamping force and maintain it during the joint's lifetime. The designer of leak-tight flange joints, especially large flange joints, must stipulate for each joint (1) the properties of the gasket,¹⁷⁻²⁹ (2) the design

method,³⁰⁻⁴² (3) the correct bolts,⁴³⁻⁴⁴ (4) the surface finish,⁴⁵ and (5) the assembly procedure to ensure that the required assembly stress is achieved.⁴⁶⁻⁵⁴

A flanged joint has several elements: the flanges, the bolts, the gasket, and the wall of the pipe or vessel. To prevent leakage, a certain level of contact stress must be maintained between the gasket and the flanges. This contact stress will vary with time due to stress relaxation in the gasket, changes in the fluid pressure and temperature with associated flange distortions, and creep in highly stressed regions of the bolts and the flanges. The net effect is a decrease in the gasket contact stress and less-effective joint seals. To maintain the necessary contact stress, elastic strain energy has to be stored in the joint system. This storage is usually in the bolt stretch, the elastic compression of the gasket, and the flange deflection. The bolts are usually the main source of stored energy but a significant amount is incorporated in metallic spiral wound types of gaskets.

Over the last ten years, there has been a major effort in the development of spiral wound gaskets, understanding the relationship between joint loading stress and the leak rate, and the improvement of joint assembly techniques. The major work in these areas is summarized in the following sections.

5.1 Gasket Types

The desired characteristics⁴⁵ in a gasket are the following: the surface layers should be plastic; the internal structure should be elastic; stress relaxation should be low; under compression, time dependent creep should be low and unaffected by temperature changes; the material should not be porous in the compressed state; the physical properties of the material should not be degraded by the fluid; the material should not easily tear or rupture; and the material should be readily available.

The gasket which meets these requirements for the HWR-NPR is a spiral-wound gasket with an expanded graphite filler and inner and outer stainless steel retainers. These gaskets have a large tolerance for flange surface irregularities;²⁸ have very good flexibility, which compensates for modest flange separation; have excellent operational characteristics; are usable at high temperature and high pressure; and are highly resistant to radiation. The main disadvantage of this gasket is that a medium bolt load is required. This type of gasket has been developed in the U.S.,^{18,21} England,¹⁹⁻²⁰ and France.²⁴⁻²⁶ One variety is being used at Savannah River²⁷ as replacements for asbestos gaskets.

Workers at Oak Ridge National Laboratory²⁸ determined the advantages and disadvantages of other types of gaskets: metal welds, elastomer O-rings, Belleville spring type gasketed joints, solid metal gaskets with a thick cross section, and hollow metallic O-rings. These advantages and disadvantages are summarized below.

Three types of weld joints are shown in Fig. 2: omega (circular around joint area), canopy (arc), lip (small weld at an edge). A canopy or omega weld uses two welds to seal the joint. A weld joint provides a leak-tight seal. However, weld joints require the use of a design

OMEGA

CANOPY

LIP

Fig. 2. Three Types of Weld Joints: Omega, Canopy, and Lip²⁸

which allows replacement of the weld five times; require qualification of the welding procedure and the welder and the testing of full-size mockups; have no strength loading; are expensive; require routine inspections; have defects which can increase permeability comparable to other seals; can have internal leakage; and are severely affected by neutron irradiation²⁹ and tritium retention, both of which cause a buildup of helium.

The perceived advantage of elastomer gaskets is that a low bolt load is required. However, elastomers have a low temperature limit; are not tolerant of radiation; have properties which change rapidly over time, especially in a tritium environment; and may have their formulation changed or discontinued.

Belleville systems are reusable by replacing an internal gasket, are light-weight joints, are usable at high temperature and high pressure, and are resistant to radiation. Their disadvantages are that critical tolerances and critical alignments are needed.

Solid metal gaskets are usable at high temperature and high pressure, are resistant to radiation and extreme temperature shock, and require low or moderate bolt loads. Their disadvantages are that fine tolerances and metal hardness are required and self-welding to the flange can occur.

Hollow metal O-ring are usable at high temperature and high pressure, are resistant to radiation, and require a low bolt load. However, stress relaxation of the metal can impair the seal, and large diameter O-rings are difficult to install.

The material which is the best substitute for asbestos in gaskets and valve packing is expanded graphite. The physical form and the properties of expanded graphite are presented.

Expanded graphite is a distinctive material with unique properties¹⁷⁻²¹ that make it a valuable material for packings and gaskets. Expanded graphite is a mechanical interlocking of worm-like expanded flakes with considerable closed porosity. Its compressibility can be changed by densification. Die-formed expanded graphite has a density of 1.4-1.6 Mg/m³ (60-70% theoretical density). The material has a high heat resistance and a negligible relaxation to sealing stress. As a filler in spiral-wound gaskets, this material requires inner support rings to prevent radial deformation of the inner windings during compression.

The manufacturing process for producing expanded graphite involves multiple steps. A mineral, like limestone containing 5% natural graphite, is treated to produce graphite flakes. Then, these flakes which are 99% graphite are separated to get a distribution which is 80% 80-mesh size particles. The 1% impurity is usually silica or mica based materials. This natural flake graphite is treated with concentrated nitric and sulfuric acid to form a bisulfate compound which enters the graphite structure, a process known as intercalation. The treated flake is then washed thoroughly, which leaves large quantities of water within the graphite structure (5-15%), together with small amounts of sulfate species (3-5%) after it is dried. (The amount of sulfur left in the material can be minimized by using an intercalation method which does not employ sulfuric acid.)

The intercalated graphite flake is then gently heated to force the remaining water to vaporize and push the graphite layers apart to give an expanded or exfoliated graphite. The degree of exfoliation achieved is closely related to the intercalation method used and the heating conditions. By heating the flakes to 1000°C in one second, a well exfoliated graphite is produced. Since graphite oxidizes at temperatures >500°C, the exfoliated graphite has to be rapidly cooled. Weight losses to oxidation are 5-10%, and water losses are up to 15%.

The degree of exfoliation is measured by the bulk density, which can be as low as 2-3 kg/m³. This low density material appears worm-like since the expansion takes place only in the "c" direction of the flake. By applying pressure to this worm-like material, the graphite is consolidated to form a coherent flexible structure in the form of sheet. These sheets can have a tensile strength of 6-10 MPa. The ultimate strength depends on the control employed during the intercalation process and the exfoliation process. The properties of the graphite sheet can be markedly improved by altering the purity of the starting flake, the source of the flake, the flake size, the reaction time in the acid, and the washing/drying process.

Some of the properties of expanded graphite, under the trade name Graphoil, have been measured by Union Carbide,¹⁸ which had the original patents on this material. Some of these properties are shown in Table 6 for Union Carbide's GTA grade sheets. Since these sheets have a density of 1.1 Mg/m³, Table 7 shows the qualitative effect of an increase in the expanded graphite's density on its physical properties.

Table 6. Room-Temperature Properties of GTA^a Graphite Sheets

Property	Quantity	ASTM Method
Bulk Density, Mg/m ³	1.1	C-559
Tensile Strength, MPa	5.2	F-152
Young's Modulus, GPa	1.4	---
Compressive Strength, MPa	165	C-695
Gas Permeability (Darcy's)	<10 ⁻⁵	C-577
Friction Static Coef.	0.05-0.2	---
Specific Heat, J/kg•K	711	--- ^b
Thermal Conductivity, W/(m•K)		Fitch Method
Length & Width	140	
Thickness	5	
Coef. Lin. Therm. Exp., 10 ⁻⁶ m/m/°C		---
Length & Width	0.4	
Thickness	27	
Electr. Resistivity, 10 ⁻⁶ ohm•m		C-611
Length & Width	8	
Thickness at 0.69 MPa	15	
Carbon Content (min), wt%	99.5	C-571
Ash Content (max), wt%	0.5	C-561
Sulfur Content (max), ppm	700	C-816
Leachable Cl/F (max), ppm	50/50	---

^aStandard sheet produced by Union Carbide. Other sheets available with density of 1.4 Mg/m³.

^bSpecific heat from Union Carbide's Carbon and Graphite Handbook.

Table 7. Effect of increase in Graphite Sheet Density on Physical Properties

Property	Effect
Tensile Strength	Increase
Young's Modulus	Increase
Compressibility	Decrease
Gas Permeability	Decrease
Thermal Conductivity	Increase
Electrical Resistance	Decrease
Recovery	Increase
Flexibility	Decrease
Hardness	Increase
Abrasion Resistance	Increase
Oxidation Rate	Decrease

The friction coefficient of flexible graphite was measured in a shear test by Union Carbide.¹⁸ On a metal surface with a 63 RMS finish and a preload of 34.5 MPa for 30 min, the friction coefficient was 0.2. Using the same test fixture with the residual graphite embedded into the metal surface, the friction coefficient was reduced to 0.09 when the preload of 34.5 MPa was held for five days. In a test with graphite sheet held against stainless steel with a 0.055 MPa load, the friction coefficient was 0.05.

The properties of graphite are as follows: (1) low coefficient of friction (0.1), (2) self-lubricating, (3) no binders, resins, fillers, (4) impermeable to gases and liquids, (5) flexible yet no cold or hot flow characteristics, (6) corrosion resistant, (7) resistant to temperature changes, (8) high degree of anisotropy for electrical and thermal conductivity, (9) capable of operating at high temperatures in reducing or inert environs, (10) chemically resistant (pH 1-14), (11) nuclear grade with leachable chloride <50 ppm, and (12) resistant to radiation.^{18,22}

The change in mechanical properties of expanded graphite sheet after 170 Mrad of gamma irradiation is shown in Table 8. Exposure to 170 Mrad is equivalent to 100 years of exposure in the nuclear reactor environment, an environment in which properties of all materials undergo degradation. Therefore, the percent changes in the mechanical properties of the graphite sheet (Table 8) are not statistically significant.

Table 8. Mechanical Properties of Graphite Sheet after Irradiation^a

Property	Control	Irradiated	% Change	ASTM Test
Ultimate Tensile Strength, kPa	5030	5720	+14	F152-72
Elongation at Fracture, %	1.8	1.3	-28	F152-72
Creep Relaxation ^b (Room Temp.), %	4	5	+25	F38-80
Compressibility, %	51.7	51.3	- 0.8	F36-81
Recovery, %	22.9	23.1	+ 0.9	---

^aAfter 170 Mrad gamma exposure from cobalt source in air. All specimens at 100 °C for 24 h.

^bSpecimens under load for 1 h.

5.2 Design Procedure for Flanges

A bolted joint functions as an elastic clamp. The behavior of this bolted clamp is complex in that the relative stiffness of the gasket, the flange, and the bolts are all important. The most widely used design procedure for joints is a long-established ASME code method (ASME 1985/Appendix S). However, this procedure focuses on bolt stresses, not gasket leakage, and uses two arbitrary gasket factors, y and m , to size bolting. The factor y is defined as the minimum gasket seating stress needed during pretightening to ensure proper gasket seating to close gasket porosity and ensure conformity with the flange face. This is not the required assembly stress to seal the joint. The factor m is defined as the stress over twice the effective gasket area divided by the internal pressure which must be maintained during operation conditions. Gasket stresses associated with tabulated values of y and m are for bolt sizing purposes only and are not sufficient to prevent leakage. The values tabulated in the ASME code are related by

$$y = 180 (2m - 1)^2 \quad (6)$$

Leakage is only reduced progressively with gasket stress. Thus, there is always some level of leakage. What a designer needs to know is what leak rate is acceptable, and then design the joint so that an appropriate stress can be applied to achieve that leak rate.

Under BHRA sponsorship³³⁻⁴¹ a design procedure has been developed which relates leak rate to compressive stress on the joint. This method is discussed below.

The joint assembly stress, S_A , must be chosen so that the gasket stress under working conditions (internal fluid pressure, p_i) is sufficient to keep leakage at an acceptably low level (specified by the design). The value of S_A is greater than y and is determined by the joint dimensions, the fluid pressure, and the relative stiffness of the gasket, the bolts, and the flanges. Several factors need consideration: gasket response, flange distortion, initial bolt-up condition, effects of thermal pressure, and thermal effects.

Since gaskets exhibit nonlinearity and hysteresis in plots of both stress-versus-strain and stress-versus-leakage,³⁴⁻⁴⁰ these relationships must be used to specify the assembly stress, S_A , needed for a given type of gasket under the expected operating conditions. Most information on gasket leakage, L_R , has been presented in the form of log/log plots of gasket stress, S_G , versus tightness, T_p ,

$$T_p = p/(L_R)^a \quad (7)$$

Here T_p is an inverse leakage parameter defined as the ratio of fluid pressure, p , to mass leakage, L_R , raised to the power a . The value for a is in the range 0.5 to 1 depending on the fluid.

Two types of tests are done to determine assembly stress: Part A, which simulates "seating" conditions, and Part B, which simulates "operating" conditions.

In Part A, the bolt load, S_G , is incrementally increased from S_N to S_A , the assembly gasket stress. At each incremental increase in the bolt load, the gasket deflection, D_{GN} , is measured, and the leak rate, L_R , is measured at several internal pressures, $p_i = p_1, p_2$, and p_3 (S_1 is adjusted to remain constant during these tests.) Before each increase in the bolt load, p_i is reduced to zero. For Part A data, a well-defined "knee" exists at K , indicating an improved joint performance above a certain gasket stress level, S_K . Beyond S_K , tightness increases more rapidly with increasing stress, indicating improved sealing performance. Similar knees have been seen with metal gaskets.⁴² The S_K value is lower when the surface finish of the flanges is rougher.

In Part B, decompression and recompression tests produce hysteresis loops, which involve little variation in the gasket deformation. Since the tests are carried out under stress-controlled conditions, cyclic creep causes a small increase in the gasket deflection with each cycle. Recompression within the previous maximum load, S_A , gives rise to an almost linear stress-deflection relationship. The tests start with the gasket bolt load at S_A and $p_i = p_{max}$. First, S_G is decreased at regular intervals, and D_{GN} and L_R are measured until the limits of leak detection are reached. Then, S_G is increased at regular intervals to S_A , with D_{GN} and L_R being measured at each interval. This sequence is repeated at each of the internal pressures, p_i .

A generic example of the results of Part A and Part B tests⁴⁰ is shown in Fig. 3, with helium as the internal fluid within an asbestos-filled spiral wound gasket. The "knee" developed in Part A tests is readily seen. In Fig. 4, data for Part A tests are shown for lines of constant gasket stress.

Fig. 3. Generic Example of Results of Part A and Part B Tests for Spiral Wound Gaskets⁴⁰

Fig. 4. Data for Part A Tests for Lines of Constant Gasket Stress⁴⁰

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One set of Part A and Part B tests was done for 304 stainless steel spiral wound gaskets with graphite filler.³⁴ For a rated 272.4 kg (600 lb) gasket with the dimensions 120.7 mm ID x 146 mm OD x 4.37 mm thick, Fig. 5 gives log/log plots of gasket stress versus tightness factor and gasket deflection at room temperature with nitrogen as the test fluid. The value for a is 0.58. These plots indicate the type of tightness possible with these types of gaskets. Qualification tests are needed on representative spiral wound gaskets with expanded graphite filler to provide designers with the characteristics of these gaskets.

The definition of the symbols in Fig. 5 is as follows:

- A_1 = the curve for $S_G = S_1$, etc.
- B_2 = the curve for $p_i = p_2$, etc.
- S_G = gasket stress using full gasket area
- S_A = assembly gasket stress = S_G in Part A
- S_K = Knee stresses in Part A
- S_K' = Knee stresses in Part B
- S_N = Intermediate gasket stresses for stress N
- D_{GA}' = Normalized gasket deflection corresponding to S_A , etc.

c , d , and f are the powers determined from the relationships

$$\begin{aligned}
 S_A/S_K &= (T_{pA}/T_{pK})^c \text{ for } S_A \leq S_K \\
 S_A/S_K &= (T_{pA}/T_{pK})^d \text{ for } S_A \geq S_K \\
 S_G/S_A &= (T_p/T_{pA})^f \text{ for Part B data}
 \end{aligned}$$

Fig. 5. Stress Versus (a) Leak Tightness and (b) Gasket Deflection for 304 Stainless Steel Spiral Wound Gaskets with Graphite Filler (272.4 kg)³⁴

Copyright 1981. British Hydromechanical Research Association. Proc. of 9th Conf. on Fluid Sealing, April 1-3, 1981, Noordwijkerhout, Netherlands, "Observations and Status of PVRC Gasket Test Program," A. Bazergui, G. F. Leon, J. R. Payne. Reprinted with permission.

k, m, and n are constants determined from experimental ratios

$$\begin{aligned} k &= (S_A - S_K') / (D_{GA} - D_{GK}') \text{ for } S_A \leq S_K' \\ m &= (S_A - S_K') / (D_{GA} - D_{GK}') \text{ for } S_A \geq S_K' \\ n &= (S_G - S_A) / (D_G - D_{GA}) \text{ for Part B data} \end{aligned}$$

Because log/log plots of T_p versus S_G are difficult to use for design, Nau⁴¹ has suggested that these data be replotted as log/linear plots of L_R/p^α versus S_G , where α is the reciprocal of a. For cases that Nau has examined, the suggested plots show as good a correlation of the variables as do the log/log plots. An example of this type of graph is shown in Fig. 6. In this figure, S_M is the required gasket working stress, and GSD is the gasket stress decrement due to bolt load and gasket reaction to the fluid end-load ($S_A - S_M$). For the case of no flange rotation, GSD is represented as

$$\text{GSD} = -(k_g / (k_b + k_g)) P / A_g \quad (8)$$

where k_g and k_b are the stiffness of the gasket and the bolt, respectively; A_g is the effective gasket area; and P is the fluid end-load, defined as

$$P = \pi (D_f)^2 p / 4 \quad (9)$$

where D_f is the effective diameter over which the fluid pressure acts. The stiffness of the gasket and the bolt are, respectively,

$$k_g = E_g A_g / t_g \quad (10)$$

$$k_b = E_b A_b / L_b \quad (11)$$

where A_b is the total effective bolting area; E_g and E_b are the unloading modulus of the gasket and the Young's modulus for the bolts, respectively; t_g is the gasket thickness; and L_b is the effective bolt length.

Flanges are assumed to be rigid, but various deformation modes occur in service. In code-designed flanges, axisymmetric ring-mode rotation is normally the most significant deformation as far as the gasket is concerned, with other bending modes being ignored. The rotational stiffness of the flange, k_ϕ , is computed by finite element analysis,

$$k_\phi = 4 \pi E_f I_f / D_f \quad (12)$$

where E_f is the Young's modulus of the flange, I_f is the second moment of the area of the flange cross-section about a diametral axis through the centroid, and D_f is the flange centroid diameter. In addition, rotation produces axial displacement of the bolts and the gasket. The expressions for these parameters are not simple, and they are not dealt with in this report.

For the case of no flange rotation, a logical design procedure is to (1) determine S_M for an acceptable leak rate, L_R' , from test data for the gasket type being used, (2) evaluate the bolt

stiffness, k_b , from bolt data, and (3) evaluate gasket stiffness, k_g , by using stress-strain data for a given gasket type.

If the bolting generates stress S_A when the joint is initially assembled, then when the joint is pressurized, the gasket stress will fall to S_M , and the working leak rate will be L_R' . In practice, correction factors may have to be incorporated to allow for inaccuracies in achieving the target bolt load and the effects of nonuniform gasket stress due to flange rotation.

Fig. 6. Gasket Leak Rate Chart for Given Gasket Type⁴¹

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5.3 Bolt Design

Bolts behave as springs and hold the joint together.⁴³⁻⁴⁹ When a bolt is preloaded, a stress is induced in the bolt. Because bolt materials are elastic, a bolt undergoes a slight increase in stress as the working load is increased. As a result, the bolt stretches. If the amount of stretch is greater than the bolt's elastic limit, the bolt yields plastically and takes a permanent set, which results in a loss in the preload stress. Even if the bolt's elastic limits are not exceeded, the induced bolt load changes with time, primarily because a material at constant stress below its

yield strength undergoes a permanent change in the direction of stress. As a result, the clamping force decreases with time.

An important factor in assuring an adequate preload on a bolt is the effect of manufacturing heat treatments on its relaxation properties. In Fig. 7, relaxation curves are shown for bolts and nuts made from AMS 6304 and tested at 482°C for 50 h.⁴³ For bolts which had been heat treated by conventional quench-and-temper methods to a tensile strength of 986 MPa (145,000 psi), the bolt preload decreased from 306 MPa to 184 MPa (44,000 to 27,000 psi). If the same material was heat treated by normalized and tempered methods to 986 MPa (145,000 psi), the bolt preload decreased only to 252 MPa (36,000 psi). If the material was heat treated to a tensile strength of 1360 MPa (200,000 psi), the bolt preload decreased from 420 MPa

AMS6304 BOLTS & NUTS, TESTED 482°C

Fig. 7. Bolt Relaxation Curves for Different Pretreatments⁴³

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to 320 MPa (60,000 to 46,000 psi). Thus, with this latter heat treatment of the bolts, a clamping force of 320 MPa (46,000 psi) is left after 50 h of relaxation. If the joint required this clamping force, one should specify the latter heat treatment for the bolts.

In addition, the design and shape of the bolt thread⁴³ have a significant effect on the bolt's relaxation properties. A stud with "square root" threads (threads which descent steeply into the bolt material) lost 50% of its clamping force in 20 h, with the curve still descending. However, a bolt with a large "radiused root" (threads with a less steep descent) lost only 36% of the preload after 35 h and the curve had stopped descending. The residual clamping force for the two cases was 238 and 306 MPa (35,000 and 44,000 psi), respectively.

An additional consideration for bolts and other fasteners is their accreditation. The ASME introduced a fastener accreditation program (FAP) on May 2, 1990⁴⁴ because of the magnitude of the problem in counterfeit and substandard fasteners. This program is intended to provide end users a means to rate suppliers and to provide them assurance of quality control and traceability of sources. An ASME document has been prepared that sets forth the requirements for engineering drawings, tooling, production planning and lot control, procurement, raw material control and traceability, personnel training, and other quality control elements. The documentation of companies desiring accreditation will be evaluated against this ASME document. The certificate of accreditation is for three years, during which time there will be at least two unannounced ASME audits.

5.4 Surface Finish

Surface waviness and tolerances increase with size and result in changes in seal compression and performance. Waviness may also result from post-machining stress relaxation or the influence of other structural components on assembly. When such effects can be quantified, they can be regarded as changes in compression and assessed.

The characteristics of a compressed surface also depend on the machining technique used to finish it. No commercial machining process produces a regular surface; there is always a probability of an occasional large peak-to-valley height forming a communicating path across the seal. It is impossible to distinguish between the effects of damage and large peak-to-valley heights due to machining effects.

The distribution of peak-to-valley heights across a surface can be approximated by a simple model,¹³ in which

$$n_h = N_0 \exp(-h/\sigma) \quad (13)$$

where n_h is the number of surface asperities of size h , N_0 is a constant, h is an equivalent hole dimension, σ is a constant in the hole size distribution function. If the flow through holes of size h is assumed to be laminar but compressible, then

$$Q_h = n_h K (h^4/L) \quad (14)$$

where K is a constant. The total flow through all holes is

$$Q_t = kN_0 \sum [\exp(-h/\sigma) h^4/L] \quad (15)$$

The peak-to-valley distribution of a highly compressed abraded surface is shown in Fig. 8.¹³ Equations 14 and 15 are plotted in Fig. 9¹³ in dimensionless form and show that the flow across the seal is dominated by that through a few, large defects. Thus, slight changes in compression can change the leak rate considerably. The variation of seal leakage with an elastomer seal's compression is shown in Fig. 10⁶ for seals produced by different machining processes. The term R_a in Fig. 10 is the arithmetic mean deviation of the surface in units of μm .

Fig. 8. Peak-to-Valley Distribution of a Highly Compressed Abraded Surface¹³

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Flange surface finishes of 0.5 to 10 μm R_a are used. The smoothest finishes are for precision joints with all-metal gaskets. The mating surfaces of flanges in bolted joints are often distorted and sometimes deeply scored. Flange waviness amplitudes of 0.2 mm (peak-to-valley) have been measured on commercial flanges.⁴⁵ When these types of flanges are bolted together with a gasket between the faces, the compressive stress on the gasket is not uniform around the circumference. The stress is highest in regions corresponding to the peaks. When the fluid pressure reduces the compressive stress on the gasket, the low stress regions, the valleys, will rapidly approach the condition in which the residual stress is insufficient to prevent leakage. Tightening the bolts will only help if the flanges are rigid enough to transfer the force to the regions of the gasket which require it, and if the mechanical properties of the gasket can allow

this increase in compression. If these conditions do not apply, then the flanges will be distorted further or the gasket will be over-stressed locally and crushed.

Fig. 9. Effect of Defect Distribution on Flow across Seal¹³

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5.5 Joint Assembly

Any effort to develop a flange seal with a low leak rate must take into consideration joint assembly to ensure that the required clamping force is achieved but does not exceed design specifications. During assembly, most of the applied torque overcomes nut-to-joint friction (50%) and thread friction (40%). Thus, only 10% of the applied torque is translated into clamping force.

Several new assembly technologies are emerging to reduce problems with bolting. These include ultrasonic control, tension control bolts, and torque-turn control. The advantage of the first technology is shown by the following experience.⁵⁰

Fig. 10. Variation of Seal Leakage with Seal Compression for Different Machining Processes⁶

Copyright 1981. British Hydromechanical Research Association. Proc. of 9th Conf. on Fluid Sealing, April 1-3, 1981, Noordwijkerhout, Netherlands, "High Integrity Static Elastomer 'O' Ring Seals and Their Performance through Thermal Transients," T.C. Chivers, A.F. George, R.P. Hunt. Reprinted with permission.

During assembly of a large diameter flange on a heat exchanger, an ultrasonic extensometer was used to monitor the variation in the bolt loads. The load on individual bolts (Fig. 11a) was random after three tightenings with calibrated torque wrenches. With the input from an extensometer, the required bolt stretch to obtain the desired bolt load was achieved for the flange (Fig. 11b). When the bolt loads on the joint were examined after two years of operation at elevated temperatures, the mean bolt stretch had decreased about 30% (Fig. 11c) but the individual load variation was minor. Since no visible leaks had been detected, the joint was not retightened.

Fig. 11. Load on Individual Bolts (a) after Three Tightenings with Calibrated Torque Wrenches, (b) with an Extensometer, and (c) after Two Years Operation⁵⁰

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During this joint's assembly, the load on a given bolt decreased as adjacent bolts were tightened. This was identified as local elastic interaction, and its magnitude is determined by the flexibility of the flange; this interaction could lead to very large variations in each bolt load. For a flexible flange, the elastic interaction is low; thus, the tightness of a bolt three bolts removed from the tightened bolt is not affected. Second, major variations in each bolt's coefficient of friction were also found; these variations correspond to major differences in the torque needed to achieve the required bolt load. Third, galling could be detected with the extensometer; minor galling produced a stick-slip phenomenon, which resulted in a sudden increase in the bolt stretch, producing very high peaks in Fig. 11a.⁵⁰

The designer of seals for the HWR-NPR should remember that (1) a uniform bolt load is as important as the mean bolt load in joint assembly, since for the same quality seal, a flexible flange requires a more uniform bolt load than a stiff flange; (2) the mean bolt load must be achieved to attain the initial gasket compression needed; and (3) the flange stiffness will determine the type of equipment needed to control bolt load during assembly, if the correct gasket is used and the bolt type is adequate. If flange rotation is <0.12 degrees, then following good assembly procedures using calibrated torque wrenches is adequate to achieve the required seal. If the flange has a rotation of ≤ 0.15 degrees, then calibrated torque wrenches and an ultrasonic extensometer are required to achieve the required seal.

Ultrasonic methods to measure bolt tension are still under development. According to Bickford,⁴⁶ some practical problems to be overcome are that accuracy is poor for short bolts and at low stress levels (173 MPa, 25,000 psi). Also, obtaining an accurate measurement is difficult because (1) bolts have a nonuniform tensile stress, residual manufacturing stresses, anisotropy, and nonuniform variations in bolt properties; (2) there are structural changes in the bolt surfaces; (3) the velocity of the ultrasonic shear wave is affected by the bolt's tensile stress; (4) there are problems with the coupling of transducers to a bolt; and (5) measurements can be affected by the position of the transducer. Because of these multiple difficulties, this technique requires additional refinement.

To bolt the joint on a reactor head, the French⁵³⁻⁵⁴ have a stud bolting robot that can simultaneously bolt several of a reactor's studs. They also use remotely operated grinding/polishing machines that clean the grooves in the closure head and the surface of the reactor vessel flange prior to the bolting operation. These machines automatically travel around the flange and use a vacuum to eliminate dust. Grinding automatically stops if the machine stops.

To achieve the bolt loads required to obtain seals with low leak rates, the designer must specify the bolt tension control strategy to be used in the assembly operation and must also specify that simultaneous bolt tightening is to be used to minimize elastic interactions.

6. ELASTOMER SEALS

The degrading effects of both gamma and tritium irradiation on the lifetime of elastomer O-rings is an important parameter since elastomers may be used to provide dynamic seals for fuel tubes, safety rods, and control rods at the reactor head of the HWR-NPR.

For most elastomers, 10 Mrad of radiation produces insignificant effects but rapid degradation occurs in all elastomers at 1000 Mrad. Atomic Energy of Canada, Limited (AECL),⁵⁵⁻⁵⁶ UKAEA,⁵⁷ Westinghouse at the Savannah River Site (W-SRS),⁵⁸⁻⁵⁹ and Los Alamos National Laboratory (LANL)⁶⁰⁻⁶³ have tested various types of elastomers to determine which has the best performance in tritium service.

Since the early 1960s, the effect of gamma irradiation on elastomers has been examined.⁶⁴⁻⁶⁸ However, because tritium is incorporated into the elastomer structure, the effects of tritium on the elastomer are more crucial in the HWR-NPR environment. Elastomers are bound together by covalent bonds that are easily disrupted by the addition of radiation energy. The effect of irradiation for a given material depends upon the chemical structure of the material. Both radiation-induced cross-linking and bond breaking lead to changes in physical properties. Cross-linking can change a material which is soft or rubbery to one which is a hard glass. Scission produces a greater number of chain ends, which leads to weakening and embrittlement of the elastomer, reducing its impact and tensile strength. Materials which soften after irradiation not only lose tensile strength but also have major gains in their compression set. (The latter denotes the degree of permanent deformation of the elastomer after a given time at a given temperature under a given stress.)

Other parameters which determine the effect of radiation are the degree of crystallinity, dose rate, presence of cross-linking agents and other additives, and O₂ concentration. Oxygen produced from radiolysis of water can cause increases in the scission rate.

Besides the effect of radiation, elastomers can have a volume change of 1-10% when immersed in water, which could change the behavior of seals within the HWR-NPR.

6.1 Long Term Deformation

The resistance of elastomer materials to long-term deformation is important for seal design. There are three techniques which have been used to measure this important property: compression set, compression set relaxation, and diametral compression set.

A correlation between these tests was established by British workers⁶⁹ for three elastomer materials: fluorocarbon rubber, low compression set fluorocarbon, and silicone rubber. Deformation data were plotted in a log/log relationship to aging time, and a linear region was extrapolated to a 100% value to give an aging time defined as the failure time, which is not the "true failure time." However, since this extrapolation errs on the side of safety, it is useful for design purposes. Failure times were found to fit an Arrhenius relationship. Arrhenius plots are shown in Fig. 12 for silicone rubber and low compression set fluorocarbon. Materials performance for these three tests can be compared by such figures. Parallel lines for the different tests imply a homogeneous aging mechanism of constant activation energy, regardless of test method.

The compression set test consistently gave a longer time to failure than the other two tests, but the other two tests are closer to service conditions, one as a face seal and the other as a diametral seal. Factors such as O-ring dimensions, temperature of conditioning before

Low compression set fluorocarbon rubber
O-rings

Silicone rubber O-rings

Fig. 12. Arrhenius Plots Comparing Compression Set to Two Other Techniques for Silicone Rubber and Low Compression Set Fluorocarbon⁶⁹

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measurement, and access of air during aging are all factors which can affect the results of any of the three tests. Thus, relevant aging conditions in service may not be faithfully reproduced by any of these tests.

Compression stress relaxation has been advocated in Europe to replace the compression set.⁷⁰ Compression stress relaxation measures, as a function of time, the decrease in resisting force exerted by an elastomer held under constant compressive strain, usually 25%. Initial strain and counterforce are measured by strain gauges. A cylindrical sample is compressed to the specified 25% strain within 30 s. The initial compressive measurement is made 30 min later, and subsequent measurements are taken at specified time intervals over 166-168 h. The compression set relaxation ratio, R_t , is calculated from

$$R_t = 100(F_0 - F_t)/F_0 \quad (16)$$

where F_0 is the initial counterforce, and F_t is the counterforce after 166-168 h. This test has been recognized overseas by ISO Standard 3384 but is not accepted in the U.S. because military specifications and ASTM D-2000 specify the compression set as defined in ASTM D-395.⁷⁰

Compression set is defined as the percentage displacement not recovered 30 min after removing the compressive load.⁵⁸ Compression set in elastomer seals results in a permanently distorted shape that is associated with stress relaxation. This change leads to an increase in leakage and forms a basis for pessimistic assessments of water leak rates from elastomer seals. Experiments show that it is relative movement, found during thermal cycling, which produces the greatest changes in leakage. If compression set occurs at a high temperature, and relative movements occur as the temperature is reduced, then sufficient elastomer elasticity may not be retained to cope with dimensional changes. Thus, large variations in leak rates could occur.⁵

6.2 Characterization of Ethylene Propylene

The material of choice for tritium service is ethylene propylene.⁵⁵⁻⁶² However, a variety of materials fall under this name; characterization work has been done with several of them, which all have slightly different properties. Ethylene propylene terpolymer (EPT)* is a copolymer of three monomers; the third monomer allows easier vulcanization than is possible with fully saturated ethylene propylene copolymer (EPR).** The former is similar to high styrene but has an increased oxygen and ozone resistance and has excellent steam and hot water resistance.

The most extensive tests were done in England by Blakeston et al.⁵⁷ with EPR, including long-term endurance tests with large seals in a reactor. The English requirements for the elastomer were that (1) seals could be made or released via remote operation; (2) seal designs could use two O-rings; (3) the material should be chemically stable in water with a pH 6-7,

* Also referred to as EPDM.

** Also referred to as EPM.

conductivity $<20 \mu\text{S/mm}$, and $\text{Cl} < 0.1 \text{ g/ton}$ of water; (4) the seal should operate reliably for five years; (5) the seal should survive in a gamma radiation field of $4 \times 10^5 \text{ rad}$ at a dose rate of 10 R/h ; (6) the seal should not adhere to metal surfaces; and (7) the seal material should not release halides to the reactor.

The change in compression set for EPR in air as a function of time and temperature is shown in Fig. 13. The material's compression set increases rapidly at temperatures greater than 100°C . The change in compression set of EPR in pressurized water (100°C and 6.5 MPa) as a function of O_2 content is shown in Fig. 14. The importance of the need to minimize the O_2 content of the water is readily seen. Blakeson et al. alluded to the effect of elastomer formulation on compression set in noting that EPT was the most promising alternative to EPR; however, no definitive information was given.

Fig. 13. Change in Compression Set for Ethylene Propylene Copolymer in Air as Function of Time and Temperature⁵⁷

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Proc. of 8th Conf. on Fluid Sealing, Sept. 11-13, 1978, Univ. of Durham, England, "The Testing of Elastomers for Channel Sea Plugs in a Steam Generating Heavy Water Nuclear Reactor,"
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At AECL,^{55,56} ethylene propylene O-rings of two different formulations (E515-80 and E692-75) were tested for up to one year as static seals in water with a pH of 10 at a pressure of 6.9 MPa and temperatures of 177 and 232°C . In addition, radiation tests were done with a Co-60 source at two different dose rates and three different doses, in water at $\text{pH} = 7, 10$, and 10 (with dissolved $\text{O}_2 \leq 10 \text{ ppm}$). Compression set was used as an indicator of seal deterioration. The compression test results shown in Table 9 for ethylene propylene are for conditions under

which the other elastomers tested had failed. A comparison between compression set and water leak rate is shown in Table 10.

Fig. 14. Change in Compression Set for Ethylene Propylene Copolymer in Pressurized Water (100°C, 6.5 MPa) as Function of O₂ Content and Time⁵⁷

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The compound with the highest resistance to deformation in a radiation environment (Table 10) was a Parker Seal (now Norton) ethylene propylene E692-75, having a compression set $\leq 50\%$ for short-term tests. For E692-75 seals, exposure to 10^7 rad had more severe compression set effects than exposure to 177°C water for 200 h. Also, the effect on compression set of radiation at 121°C was more severe than the effect at 15°C.

A radiation-resistant elastomer to replace Buna-N rubber was desired at Savannah River Laboratory; thus, ethylene propylene E740-75 and other elastomers were tested⁵⁸⁻⁵⁹ using compression set as an indicator of radiation resistance. The compression test was done two months after exposure of the elastomers to tritium. As shown in Table 11, the compression set for ethylene propylene E740-75 in the presence of 0.07-0.08 MPa molecular tritium was 52% at 125 days tritium exposure and decreased to 74% at 225 days tritium exposure. The "radiation-induced set" is the difference between the total compression set for the irradiated elastomer and that for a control exposed to air. The tritium off-gas rate, also listed in Table 11, gives a relative indication of the solubility of tritium in each elastomer.

Table 9. Conditions for which Compression Was $\geq 50\%$ for Ethylene Propylene (E515-80 and E692-75) in AECL Tests^a

Conditions	Time, h	Dose, rad	Compression Set, %
177°C in air	20	---	>50 (only E515-80)
177°C in air	200	---	>75
15°C in air ^b	<50	7.4×10^7	>50
177°C in water, all pH's	200	6×10^7	>50

^aOther materials had failed prior to these conditions.

^bEffect was independent of dose rate, i.e., time.

Table 10. Water Leak Rates and Compression Set for Ethylene Propylene Compared to Other Elastomers

Seal Material	Leak Rate, mL/min	Comp. Set, %
Ethylene Prop. E692-75	0 at 15 cycles ^a	50
Ethylene Prop. E692-75	0 at 10 cycles	47
Ethylene Prop. E692-75	0 at 5 cycles	42
Ethylene Prop. E515-80	0 at 5 cycles	75
Viton 19357	0 at 5 cycles	61
Viton V747-75	High at 3 cycles	96
Perfluoro ECD 006	High at 1 cycle	97

^aCycled from 177°C/6.9 MPa to ambient conditions.

Table 11. Compression Set and Tritium Off-gas Rate for Ethylene Propylene and Other Elastomers

Elastomer	Total Compression Set, %		Rad. Induced Set, ^a %		Off-gas Rate $\times 10^7$, cm ³ (STP)/s
	125 d	225 d	125 d	225 d	
Ethylene Propylene (E740-75)	52	74	44	66	3.6
Styrene Butadiene (G244-70)	75	97	68	88	9.1
Kalrez Perfluoro 4084	na	85	na	46	na
Kalrez Perfluoro 1018	83	na	34	na	11
Viton B (V494-70)	88	na	42	na	11
Viton E60C (V747-75)	93	97	69	75	20
Ethylene Acrylic (A936-70)	93	97	77	83	39
Nitrile (N741-75)	97	97	90	88	14
Urethane (P642-70)	98	100	77	71	140
Silicone (S604-70)	99	na	90	na	120

^aDifference between irradiated sample and control held in air.

Since there are different types of ethylene propylene, a non-proprietary formulation of EPDM was prepared at LANL⁶¹⁻⁶² and tested to ensure that a material with reproducible properties and formulation would be available over several years. The desired properties were the following: chemical stability, long shelf-life, minimum H_2 permeation, low compression set, durometer hardness of Shore A ≤ 80 , temperature range of -55 to 75°C , and minimum elongation (100%).

Two roll mills were used to attain a compound with the dispersion needed for small diameter seals. High peroxide and high cross-linking co-agent were used to achieve a high cross-link density, which makes the material less sensitive to radiation. A vacuum post cure was used to remove residues, which contribute to further cross-linking when a seal is compressed. An acceptable material was one in which $<2\%$ of the total material was lost. No plasticizer or sulfur was added. Even with a non-proprietary formulation, the material characteristics could change from year to year since the formulations of the starting ingredients (trade names) could be changed in the future.

The compression set of the EPDM material was 5% after 400 days exposure to molecular tritium at room temperature. The maximum compression set was 10% after 70 h at 125°C . The hardness was Shore A 78; the tensile strength was 8.2 MPa; the minimum elongation was 100%. The tritium permeability was $<10^{-6} \text{ cm}^3/\text{s}$ after 10 min. After 270 Mrad gamma irradiation, the permeation decreased 15%. This result could indicate elastomer movement against the gland face and/or EPDM degradation.

Several ASTM methods measure the release of gases from elastomers, depending on the form of the elastomer (films, sheets, thick materials, coatings). However, the results from these methods for the same material from the same company can vary by a factor of three or more. Using one of these methods, Cassidy, Aminabhavi, and Thompson⁷ measured the permeability of water in EPDM and found it to be $2 \times 10^{-7} \text{ cm}^3/(\text{cm}\cdot\text{s}\cdot\text{MPa})$.

The problem associated with the continuous reformulation of elastomers by the polymer industry is highlighted in the case of another elastomer.⁶³ To obtain a species with a lower hydrogen permeability than the non-proprietary EPDM,⁶¹⁻⁶² Hydrin 100, unfilled and filled with hydrous MgSiO_3 , was examined. The deuterium permeation was 5-45 times less than that of EPDM. However, the material is no longer made because the company was sold. A material known as H-65, made by a different company, may be comparable but this has to be verified by repeating all tests.

6.3 Engineering Design of Elastomers

O-rings are designed to be deformed. A designer is interested in how much force is needed for a given deformation, since this affects the achieved leak rate and the assembly procedure for a seal. The Young's modulus varies with the amount of strain and can be related to the hardness of the elastomer material. The general relation⁴⁵ between Young's modulus (N/mm^2) and hardness (IRHD = International Rubber Hardness Degrees) is not linear, rather the relation is 1.47/40, 1.76/45, 2.14/50, 3.16/55, 4.34/60, 5.72/65, 7.16/70, 9.24/75, and 27.6/90 at room temperature. Elastomers for static seals would have a hardness of 90, those for dynamic

seals would have a hardness of 70. The lower hardness results in a lower friction force. Since temperature affects the Young's modulus, stress relaxation, and expansion of an elastomer, the prediction of an elastomer seal's performance depends on a projected time-temperature-squeeze history of a joint.

Even though the hardness properties of an elastomer are well known, an elastomer's use in a design has to be modeled.^{45,71-73} Elastomer O-rings are pressure actuated and should be pressurized in the right direction. Correct constraint geometry and pressure direction should be modeled.

The failure to model the operation of the Challenger seals was a root cause of the 1986 accident. Seal design for the O-rings in the space shuttle solid rocket booster was based on conventional practice, limited data from the manufacturer, and previous experience. The interplay of surface finish, friction, squeeze, end constraints, and time-temperature-dependent material properties on reliability was not recognized.

To analyze the behavior of the O-rings in the solid rocket booster after the shuttle accident, Metcalfe et al.⁷³ used a large displacement, incompressible, viscoelastic finite element model (MARC code)⁷⁴ and determined the O-ring squeeze during assembly of the booster case, followed by relaxation, recovery, and pressurization of the seal. The results showed the feasibility of predicting an O-ring's dynamic response to conditions, including thermal cycling and deformation and/or movement of the joint. In effect, this study⁷³ was a practical demonstration that the viscoelastic response of an O-ring to squeeze, dwell time, gap widening, friction, temperature, and pressure can be successfully modeled using the MARC code, combined with a rigorous approach to materials characterization, boundary conditions, and incremental solution procedures. The appendix of Ref. 73 contains a detailed theoretical basis of the approach.

7. SEALS IN VALVES

Valve leakage is a major contributor to reactor plant contamination and maintenance. The sources of valve leakage in order of importance are (1) stem packing leakage, (2) seat leakage due to chemical buildup, (3) body-to-bonnet joint leakage, (4) valve damage due to cavitation, and (5) excessive seat closure loading.

Valve stem packing is the most significant source of leakage since packings consolidate and wear with time, resulting in insufficient elasticity to accommodate dimensional variation. The factors which control stem packing leakage are the composition of the packing material and the presence or absence of a dynamic system that provides the needed packing gland pressure. Of secondary importance are the frequency of valve stem movement, the finish on the stem and the stuffing, the quality of the surfaces in the stuffing box, the clearances, and the depth of packing.

Valve stem packing leakage can be handled by providing systems to collect the leakage, replacing the packed valves with bellows valves, eliminating the packed valves, and/or redesigning the packed valves. Since collection systems can become overloaded and leak, and

bellows valves cannot be used for some applications, one must be able to redesign the packed valve.

The effectiveness of stem packing depends on the stuffing box design, the materials used for packing, the manner in which force is applied to the packing, and the finish on the stem. Workers in Canada,⁷⁵⁻⁸⁷ France,⁸⁸⁻⁸⁹ England,⁹⁰ Germany,⁹¹⁻⁹² Japan,⁹³ and the U.S.⁹⁴⁻¹⁰⁰ have sponsored work to increase the effectiveness of stem packing. Three manufacturers have worked in this area, Velan Manufacturing Co.,^{85,90} Latty International, France,⁸⁸⁻⁸⁹ and Rockwell International.⁹⁹

Lipton¹⁰¹ conducted dimensional checks on valves from different manufacturers and found a lack of standardization. Many firms do not understand what compressive stress is needed to achieve a valve packing seal with a low leak rate.¹⁰¹ Except for the data supplied by AECL,^{75,87} for valves with asbestos packing, and recently by EPRI^{94,97-98,100} for valves with expanded graphite packing, little to no information is available to manufacturers on ways to optimize the design of the packed stuffing box, which is the basis of a packed valve's seal. Work done by one of Rockwell's divisions is summarized in Ref. 99; however, this expertise appears lost since the division no longer exists.

The following sections address the usefulness of bellows valves and the design of packed valves. Some other valve developments which may be applicable to the HWR-NPR design are also mentioned.

7.1 Bellows Valves

A valve which does not have stem packing is a metal bellows valve. Closure is affected by a flexible connection between the traveling valve stem and stationary bonnet. A packing gland and/or a weld is added to provide protection in event of rupture. Failure of the bellows occurs due to fatigue induced by excess vibration at the resonance frequency of the bellows or by excess turbulence in the fluid.

The advantages of bellows valves are that they have indefinite "zero" leakage and low "friction." Their disadvantages are that stroke is limited, pressures in excess of 1 MPa require balancing, fatigue life is limited, failure can be catastrophic with little warning, large exposed surface areas can increase degradation, and age hardening can reduce fatigue life. The limited stroke introduces a practical size limitation. For acceptable fatigue life, a bellows for a 30-cm gate valve would have to be 3-m long because the bellows can only be stroked 10-15% of its free length. This size bellows valve would be expensive to fabricate. However, bellows valves can be a very useful design solutions since the majority (80%) of valves in reactors are under 5 cm in diameter.

A bellows valve should have a backup stem seal to prevent catastrophic leakage if the bellows fails. In CANDU reactors, bellows valves have redundant packed-stem seals or a weld to back up the bellows. During operation of 600 bellows valves (2.5-cm diameter) through 1975 at Pickering,⁷⁵ 11 bellows failures occurred for a mean time between failures of 560 y (MTBF = 4.9×10^6 h).

Bellows valves for the HWR-NPR should be subject to proof tests. The criterion for acceptance should be a value of 5000 for the number of failures versus the number of stem actuations. Other tests should address pressure cycling, functional flexing, vibration, resonance frequency of the bellows, and the adequacy of the shield between the bellows and the fluid.

7.2 Packed Valves

During the 1970s, the AECL and the Velan Manufacturing Co. collaborated on a valve development program.⁷⁵⁻⁸⁷ Work concentrated on three valve types which cannot be replaced with bellows valves: gate valves which have a large (25-35 cm) diameter⁷⁹; globe valves which undergo fatigue due to thermal stress; and control valves which undergo fatigue due to constant movement during reactor operation.

The AECL reviewed the theory of leakage through porous media and developed an empirical mathematical framework for relating leakage and packing friction to stuffing box dimensions and compressive stress.⁸⁴ They developed a procedure to modify conventional valves and to develop new valve designs.

The four basic findings of their program were the following. First, leakage can be predicted from the knowledge of the stress applied to a known type of packing. Second, packing materials each consolidate in a predictable manner under sustained load. Third, spring mechanisms consisting of Belleville washers can compensate for packing consolidation yet sustain sufficient load to control leakage. The spring mechanism is designed into the stuffing box to accommodate dimensional variation and maintain the packing gland pressure. Fourth, the applied packing load can be determined from the height of calibrated springs. (A spring-loaded system is designated "live-loaded packing.")

Live-loading of packing glands is used in Canada and France; their valves have been designed to optimize the use of live-loading. Canada has used live-loading with asbestos packing in CANDU reactors since 1976. The mean time between maintenance for their asbestos live-loaded packing for gate valves was 19 y (MTBM = 1.7×10^5 h). For live-loaded globe valves, the MTBF was 4.9×10^6 h. France uses a combination of graphlite stem packing and live-loading for packing glands in PWRs.

The first use of live-loaded valves in U.S. PWRs was sponsored by EPRI.^{94,98,100} Modified valves with wedge-shaped expanded graphite packing and live-loading were retrofitted into some of the Duke plants.¹⁰⁰ Due to the limited nature of the study, these modified valves were not optimized designs, but they had a significantly reduced amount of leakage. The success of the retrofit program has prompted the suggestion that all valves be redesigned to incorporate this design to achieve valves with very low leak rates. However, live-loading is not in the design specifications of any U.S. power plant.

7.2.1 Live-Loading

Live-loading is the use of a stack of disc springs, placed under the gland nuts on either side of the valve stuffing box, to exert a constant and controlled force on the valve

packing such that the axial stress remains within narrowly defined minimum and maximum values as the packing consolidates. (The distribution of axial and radial forces within the stuffing box is shown in Fig. 15.) Doubt⁸⁴ defined the minimum and maximum values through calculations in which packing material composition, valve rating, cycling frequency, and service conditions were considered. Once the disc springs are in place and the gland nuts are tightened to the calculated value, the result is a self-adjusting leak-tight seal. No further maintenance or periodic tightening of gland nuts is required.

Fig. 15. Distribution of Axial and Radial Forces within a Valve Stuffing Box.⁸⁰

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As shown in Fig. 16, live-loading reduces the slope of the force-displacement relationship for the gland loading system. Live-loading with disc or Belleville springs provides 20-25 times the energy storage capacity provided by stuffing box (elasticity) and gland follower studs alone. For a disc spring, the gland pressure is constant as the spring deflects. The advantages of springs for live-loading are that they have the capacity to absorb a large amount of energy at high loads and short working stroke, and can be easily stacked in parallel for different stack spring rates. Live-loading with disc or Belleville springs can accommodate thermal expansion, tapered stems, stem wobble, and stem-side loads.

Fig. 16. Force-Displacement Relationship for the Gland Loading System⁹⁴

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The theory of load-deflecting characteristics of Belleville springs was reviewed by AECL staff⁸⁰⁻⁸¹ for data from the alloy steels AISI 6150 and AISI 1075, as well as the precipitation hardened stainless steel 17-7 PH in the temperature range 26 to 288°C. The theoretical load-deflection relationship for a single Belleville spring is:

$$L = E dd [(h_o - dd)(h_o - dd/2)t + t^3] / [(1 - \mu^2)M(r_o)^2] \quad (17)$$

where L is the load, E is Young's modulus, dd is the spring deflection, h_o is the spring dish height (height of the base of the spring from the horizontal plane), t is the spring thickness, μ is Poisson's ratio, r_o is the spring outside radius, and M is

$$M = [6/(\pi \ln(OD/ID))] [((OD/ID) - 1)/(OD/ID)]^2 \quad (18)$$

In Eq. 18, OD is the outside diameter and ID is the inside diameter. Based on Eq. 17, the response of a Belleville spring is dependent on the geometry of the spring. By varying the dish height-to-thickness (h_o/t) ratio, different characteristic load-deflection curves are obtained, as shown in Fig. 17. The Canadians compared the observed load-deflection curves with the theoretical ones and found that close agreement between tests and theory only occurred if $h_o/t > 0.7$. For $h_o/t \leq 0.5$, the stack spring height cannot be closely estimated by theory. For $h_o/t > 0.5$ and < 1.4 , one can obtain a constant load over a large range of spring deflection. The

Fig. 17. Characteristic Load-Deflection Curves for Belleville Springs¹⁰⁰

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vertical line in Fig. 17 shows where the spring deflection equals 75% of the unloaded dish height, h_0 . This deflection should not be exceeded for live-loading applications.

For the design of spring stacks with specific spring rates, the selection of the stack configuration should be based on observed load-deflection data for a particular spring. Spring deflection must accommodate in-service packing consolidation yet maintain the required force, once initial consolidation is achieved at installation. Two springs stacked in series equals twice the deflection of a single spring. One can stack a maximum of two identical springs in parallel, with the two being nested over each other. Each spring supplies 50% of the required sealing load. Four types of spring stacking arrangements are shown in Fig. 18. In addition, load relaxation of the springs, which is <6% at room temperature but increases linearly with temperature, has to be included in spring design.

7.2.2 Expanded Graphite Packing

Most of the work done by the Canadians was done with asbestos packing or a combination of asbestos and graphite. Since expanded graphite has become the packing material of choice, obtaining data on the use of this material in packings is especially important for the HWR-NPR design.

In 1982, valve and valve-packing manufacturers, as well as nuclear plant valve users, recommended the use of graphite packing in valves.⁹⁷ A survey of Licensee Event Reports (LERs) from 1972-1980 using statistical analysis confirmed that the adoption of graphite packing significantly reduced valve stem leakage. Maintenance experience with expanded graphite as a packing material in reactor valves in Swedish and French plants has been excellent. There was no need for repacking between shut-down periods when the expanded graphite¹⁹ was correctly specified and installed. Recent development work sponsored by EPRI has been directed at determining if there is an optimum design for graphite packing.

Packing depends on axial compression, initially from a gland follower, then from the applied system pressure. This compression creates a radial loading that produces a seal between the valve shaft and the packing (Fig. 15). Packing rings that are die-formed from expanded graphite tape require 50% of the gland load of braided asbestos to seal.¹⁰⁰ Expanded graphite relaxes slightly, 5-6%, immediately after tightening, then relaxation ceases.⁸⁹ Expanded graphite when formed into a ring of $\geq 80\%$ theoretical density has very little relaxation, stability over a wide range of temperatures, good behavior in boric acid, and good behavior under irradiation.¹⁸ The graphite has to be of the highest purity to minimize corrosion, e.g., ash <0.2%, Cl <50 ppm, S <600 ppm.

In an EPRI study¹⁰⁰ three expanded graphite ring shapes -- sootblower (a parallelogram), chevron, and wedge -- were compared with a square graphite ring shape to determine which required the lowest gland pressure to achieve a good seal. Sootblower packing was difficult to fabricate and to load because of its shape. Because chevron packing expanded as it was loaded, it too was difficult to load and its edges were prone to damage. Wedge packing had a truncated apex on the outer rings to allow motion between packing rings, making it easy to

load. Early in the study, studies on sootblower and chevron packing were dropped due to the above problems.

- a) Single Spring
- b) Two Springs in Parallel (Twice the Force of a Single Spring)
- c) Three Springs in Series (Three Times the Deflection and the Same Force as a Single Spring)
- d) Three Parallel Pairs (Twice the Force and Three Times the Deflection of a Single Spring)

Fig. 18. Belleville Spring Stacking Arrangements¹⁰⁰

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A full wedge packing set was seven wedge rings (four inner, three outer), and two braided end rings. This set is equivalent to 3.5 square die-formed rings. Braided end rings act as wipers and prevent graphite particle buildup. They have no sealing capability and experience 50% initial consolidation. The four inner wedges have their bases on the stem while the three outer wedges have their bases on the stuffing box and truncated apex. This arrangement is shown in Fig. 19.

Fig. 19. Graphite Wedge Packing Arrangement in Valves⁹⁴

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During initial installation, the valve must be cycled to allow the packing to consolidate, especially the braided graphite. One needs to stroke 10 times when the valve is first installed to account for 20% consolidation and then retorque the gland nuts before putting the packing into service. The minimum gland pressure is 13.8 MPa (2000 psi) on the graphite rings. The effect on the gland load of the initial consolidation of the packing and later in-service consolidation is shown in Fig. 20. In-service consolidation is $\leq 2\%$ of the packing stack height.

For wedge packing the gland load was well below the system pressure to achieve a seal (Fig. 21). This finding implies that wedge packing uses system pressure to load packing and achieve sealing. The compliancy of wedge packing allows it to conform to thermally induced tapers in large-diameter valve stems. Table 12 gives test results in which leak rates for graphite wedge packing with and without live-loading are compared to those for graphite square packing.

Fig. 20. Initial Consolidation and In-Service Consolidation of Graphite Wedge Packing¹⁰⁰

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The use of wedge packing in place of square packing has several advantages. It provides improved leakage control during valve stroking, adjusts to the stem taper to reduce leakage, maintains the radial sealing forces over an extended period, adapts readily to dimensional variation, and is easier to install.

Packing friction is an important consideration in the design and sizing of valve motor actuators. On average, live-loading produced 80% higher initial in-stroke friction, per unit of gland pressure, than conventional loading.¹⁰⁰ The higher friction is due to the more uniform gland load that the springs apply to the packing relative to conventional gland loading.

A simple formula for initial friction, F , is

$$F = \pi f d L Y \sigma_g \quad (19)$$

where f is the friction coefficient for in-stroke or out-stroke friction, d is the stem diameter, L is the packing length, Y is the ratio of radial to axial stress in the packing, and σ_g is the average contact stress between the packing and the gland follower. This formula does not account for the stress gradient in the stuffing box. Over the long term, in-stroke and out-stroke friction is affected by temperature/diameter variations along the stem, material stress relaxation, and apparent changes in the elastic properties of the packing due to consolidation. These factors substantially reduce the long term friction compared to the initial friction force, with the result that the long-term friction for live-loading and conventional loading is comparable.

* The results obtained for the asbestos packing were with an acceptable leakage rate. The graphite packings exhibited no leakage during testing.

Fig. 21. Gland Pressure Versus System Pressure for Graphite Wedge Packing.⁹⁶ (Also plotted are results from asbestos and square graphite packing.)

Table 12. Dynamic Leakage for Graphite Packing in BWR and PWR
Modified Valves After 1500 Stroke Cycles

	Packing Type/Loading Type ^a			
	SQ/CON	SQ/L-L	WG/CON	WG/L-L
BWR				
Leak Rate, L	155	4	11	3.2
Relative Rate ^b	1.0	0.026	0.071	0.021
PWR				
Leak Rate, L	117	25	11 ^c	4
Relative Rate ^b	1.0	0.21	0.094	0.034

^aSQ = graphite square packing, CON = conventional gland loadings,

WG = wedge packing, L-L = live loading.

^bBased on 1.0 for square packing with conventional gland loading.

^cInitial gland pressure was 10.3 MPa (1500 psi), unlike live-loaded wedge, which was started at 19 MPa. Gland pressure was later increased to 19 MPa.

7.2.3 Valve Redesign

The design of the stuffing box for a valve has to be modified to incorporate live-loading. To retrofit valves, one has to first know the disc spring characteristics, determine if there is clearance for the spring stack, calculate the maximum and minimum sealing force required, and know the in-service consolidation expected for the packing. Once these factors are taken into consideration, one can modify the valve and add disc springs, disc spring guides, and elongated gland studs.

A correct fit is needed to prevent delamination of the graphite rings and to ensure that the gland load goes to ring deformation and not to radial sealing force. One needs ring tolerances within +0.013 to -0.000 cm for the ring ID, +0.000 to -0.025 cm for the ring OD, and ± 0.038 cm for the ring thickness. For the valve, the stem diameter in three locations is needed to ± 0.0025 cm, as is the stuffing box diameter and the stuffing box depth.

Rockwell⁹⁹ attempted to retrofit some of their valves with asbestos packing to valves with graphite packing and live-loading. Their valve stem diameters were 2.5-10 cm, and the stem strokes were 4.27-51 cm. Over 100 packing tests were cycle-tested over 12 months for several types of graphite. The best results were achieved with the Union Carbide form of expanded graphite. A comparison can be made between the dimensions for a valve which uses asbestos packing and one which uses expanded graphite packing. The packing volume is reduced over a factor of two and the packing width a factor of 1.3. In Table 13, the packing cross section is given for valves with 25.4-, 60.3-, and 101.6-mm stem diameters.

7.3 Other Valve Developments

Two types of valves have been developed under EPRI's auspices¹⁰² as replacements for high-maintenance (2.5-cm dia) valves: a rotating-stem, dual-plug valve and a hermetically sealed, actuated, dual-plug valve. The first valve operated well, but the second failed due to

Table 13. Cross Sections for Graphite Packings Tested by Rockwell

Stem Diam., mm	Packing Cross Section, mm	Packing Rings, No.
25.4	6.4	4
60.3	9.5	4,5
101.6	12.7	5

galling of internal parts and/or stem seizure due to a galled bushing. The latter valve was redesigned to eliminate the bushing by replacing it with a ceramic bearing.

Since movement of a valve stem could produce leaks, a microprocessor¹⁰³ can be retrofit on valves to control the movement of the valve stem and the consolidation of the packing. Also, for maintenance convenience, indicators can be designed into valves to detect maximum compression of the packing.

For check valves, which have internal leakage due to wear, several groups have examined their maintenance.¹⁰⁴⁻¹⁰⁶ The check valve failure rate is 1-2% and is caused by poor maintenance and inadequate preventive maintenance. To improve this record, the MOVATS ultrasonic diagnostic system¹⁰⁵⁻¹⁰⁶ has become an integral part of the maintenance program for these motor-operated valves. Dynamic signatures (ultrasonic trace of a valve's response) of critical actuator parameters are obtained and analyzed to identify degradations in check valve performance.

8. SEALS IN PUMPS

Two types of pumps are considered in this review, centrifugal pumps¹⁰⁷⁻¹²⁷ and canned motor pumps.¹²⁸⁻¹³⁹ Problems with seals, external or internal in these pumps, are reviewed.

8.1 Centrifugal Pumps

Frequent seal failures in centrifugal pumps used at CANDU reactors in the 1960s led the AECL to use custom-designed axial and elliptical seals and improved bearings. This resulted in pumps with seal lives in the Pickering plants of >2.85 y in 1975. For later plants, pump testing was used to verify quality control of the seals.^{108,113} At the Bruce power station, 16 primary pump sets were fully loop tested for at least 100 h, while four were proof tested for 1000 h. In addition, seal cartridges accumulated several thousand additional testing hours in test rigs. At the CANDU plants, early seal failure in the pumps was usually due to dirt and improper assembly.

Most reactor coolant pump failures¹²² in the U.S. are attributed to pump seals, followed by pump motors and mechanical components like bearings and flanges. Maintenance and operating data indicate that most seal failures are attributed to improper seal injection; crud in the system, which could be due to seal decomposition or wear; and the fretting of rubber faces in inserts and in the seal housing.

The shaft seal assembly of reactor coolant pumps in U.S. PWRs and BWRs consists of 3-4 stages of rotating face seals. Each stage includes elastomer O-rings to seal static gaps, an elastomer secondary seal to accommodate small motions between parts, and a small-gap hydraulic primary seal. Each stage operates by maintaining a very small leakage pathway between two seal rings which form the primary seal. The gap is maintained by a balance of forces (hydraulic, spring, and friction), which can be influenced by the fluid conditions in the seal cavity.

Large-diameter, high-pressure elastomer seals, which are used as secondary seals in rotary end-face seal assemblies of the main coolant pumps of PWRs and PHWRs, are susceptible to rapid damage because such seals must accommodate slight cyclic movements caused by misalignments between the inflexibly mounted end-face seals and the rotational axis of the pump shaft. The damage is characterized by a circumferential wear groove in the elastomer seal and later by wear of all areas of the seal in contact with the pump shaft. In plants not plagued with frequent seal replacement from other causes, elastomer damage is a major factor limiting the service life of coolant pump seals.

Wensel¹¹² tested three seal forms (curved-back U-cups, flat-back U-cups, and O-rings) for three elastomer materials (ethylene propylene, neoprene, and nitrile rubber) to determine their behavior during conditions of a power blackout. The damage mechanism was localized fatigue and thermal deterioration of the elastomer material in the area of the seal subjected to greatest flexure from cyclic motion. Abrasion of the seal against the pump shaft sleeve may also have been a factor. Flat-back U-cups were rapidly damaged, resulting in leakage of 20 mL/h in 48 h. The O-rings were damaged and leaked at a rate much lower than the flat-back U-cups. The curved-back U-cups, although damaged, did not leak after more than 1000 h of testing.

In later stages of the tests, ethylene propylene, neoprene, and nitrile rubber were all damaged with equal rapidity when grazed by a water jet, although the damage rate was slightly less for the ethylene propylene curved U-cup. Wensel recommended that the angular misalignment be minimized between the rotational axis of the pump shaft and the face of the inflexibly mounted seal, elastomer seals with the most susceptible shapes (flat-back U-cups and square section rings) be avoided, pump shaft sleeve and seal lip be prelubricated, the shaft be polished to lower its friction, and the sharp edges of the seal gland be polished to a 0.25 mm radius.

Temperatures of 260°C will cause degradation of elastomer seals and will affect the pressure distribution of the flow through the gap between the seal rings. Subudhi et al.¹²² recommended the use of ethylene propylene (E740-75) rather than conventional E515-80, since the former has a better resistance to high temperature. (The E740-75 also has a lower compression set in the presence of tritium.) Increased pressure could cause the hydraulic seal to open to a larger gap (bi-stable seal) or to pop completely open (unstable seal) and allow leakage.

The parameters which had the most influence on seal stability¹²¹ were inlet fluid conditions, back pressure on the discharge side (inside diameter of the seal rings), and the radial taper in the gap between the seal rings. Analysis showed that stability, although a necessary

condition for limiting flow through the primary hydraulic seal, is not sufficient to assure acceptable leak rates.¹²¹ For example, a seal with a gap convergence of 762 μm was calculated to leak in excess of 13 L/s at 15.3 MPa, even though it was operating in a stable manner. The leak rate is highly dependent on the combination of inlet fluid conditions, back pressure, convergence, and closing balance ratio* but is not readily characterized as a function of those parameters.

Other changes recommended in reactor coolant pumps¹²² were to change seal face-plate materials to silicon nitride, to provide chromium carbide clad on stainless steel for all seal rubbing surfaces, and to change from graphite-impregnated Teflon to Tetralon 720 (the former being adversely affected by low levels of radiation $\leq 10^5$ rad) for double delta (or channel) seals.

8.2 Canned Pumps

A canned motor pump¹²⁸⁻¹³⁴ is driven by a magnetic coupling between the pump and an external rotating motor. Electromagnets in the external motor drive the internal rotor, which is sheathed in a metal "can." The pumped fluid lubricates the pump's rotor cavity and the bearings within it. A major problem for a canned motor pump is the buildup of corrosion products arising from iron in the rotor. Corrosion is accelerated in an acid environment, which will be present in the NPR-HWR. The use of stainless steel rotors will lower the corrosion rate but not eliminate the corrosion process.

For a canned motor pump, leakage to the external environment is minimized since seals, which account for the majority of a centrifugal pump's leakage, are eliminated in the pump internals. (There is a gasket seal for the pump housing which can be welded closed or replaced with a spiral wound expanded graphite gasket.) However, internal leakage can occur as a minor breach of the internal "can" or as catastrophic failure. The cause of these "can" failures is the buildup of corrosion particles in the internal lubrication system of the pump or the failure of the bearings and/or the rotor due to wear. Damage to the bearings can result in rubbing of the rotor against the "can," causing failure. Since the external housing of the canned pump can be a high pressure boundary, failure of the "can" can be contained within the housing.

Compared with centrifugal pumps, canned pumps have fewer leaks, need less maintenance and repair (80% of conventional pumps' problems are seal problems), take up less space for a given size, and run quieter. However, canned pumps cost more; their efficiency is lower due to parasitic power losses (20-30%), which depend on the pump design, its size, and the application; and they can be more easily damaged than seal pumps. The power losses are 5-10% due to eddy currents in the shell and 15-30% due to the mechanical gap between the stator and rotor. The canned pumps are more easily damaged for two reasons: pipe scale or bearing wear can block small passages in the internal lubrication system, resulting in major mishap; and

*Closing balance ratio is the ratio between the closing load due to the pressure on the back side of the seal rings balanced by the hydraulic opening load due to the pressure between the seal faces. Seals with a low closing balance ratio are most likely to pop open.

running the pumps when they are dry results in worn bearings, internal rotor galling, and rupture of the can.

If canned pumps are used, one needs filters to remove particulates; a detector to signal a dry pump; a thicker can; an outer shell for secondary containment if the can ruptures; a moisture-sensing probe interlocked to an alarm and pump motor; better bearings, preferably expanded graphite bearings rather than the current graphitar bearings in Westinghouse pumps¹³⁵; and a separate clean fluid for the bearings maintained under pressure by a bellows seal.

Canned motor pumps with 3 L/s throughput are in use at the Savannah River Site (W-SRS) in the water purification area. These units are discarded if a failure occurs. This practice is not a large expense for pumps of this size¹³⁹ but would not be attractive for reactor coolant pumps. At W-SRS, the current reactor coolant pumps are large centrifugal Bingham pumps, 1575 L/s.

9. DESIGN GUIDELINES

The need to reassess common practices for the design of bolted flanges, elastomer seals, valves, and pumps has been mentioned. This section highlights some design guidelines which, if used, will result in components for the HWR-NPR coolant system with acceptable leak rates.

9.1 Flange Seals

For large diameter seals in critical locations, a finite element analysis is required to determine the rotation in the joint, which is a function of the internal pressure and the structural stiffness of the flange. The seal should be modeled full size, since deformation characteristics and contact stress are influenced by a seal's cross section and scaling from data at 0.1 MPa differential pressure could be misleading.

The ASME formulas used to compute the sealing load for a flange are not justified in view of existing data. An alternative system (BHRA development) should be used to specify compression loads for flange seals that contain a spiral wound gasket with an expanded graphite filler.

9.2 Valve Seals

The valve designer has several choices for the HWR-NPR plant. First, the number of valves can be minimized within the constraints of system maintenance. Second, bellows valves or welds can be used if fatigue is not a problem or if loss of access is allowed. Third, valves using die-formed expanded graphite packing with live-loading can be incorporated. For the latter, packing and spring combinations have to be tested so that spring stacks can be designed to assure a sustained, high gland load on the packing. This option probably will require the design of new valves. The general actions that the designer must take to design packed valves with low leak rates are summarized in Table 14.

Table 14. Designer Actions to Achieve Valves with Low Leak Rates

Fact	Action
1. Packing materials change properties with time at operating conditions.	1. Choose material which remains stable a long time.
2. Packing materials consolidate under sustained load in predictable manner.	2. Provide spring-loaded device for packing which is capable of maintaining load at an acceptable level.
3. Leakage must be controlled.	3. Control applied packing stress.
4. Valve industry has no strict standard for stem seal design.	4. Issue detailed stuffing box specification.

Four more guidelines arise from Ruggieri and Kelly.⁹⁶ First, since a maximum spring OD gives the greatest load capability over the largest amount of deflection, fewer springs are required to accommodate in-service consolidation. The radial clearance of the valve studs governs the maximum OD. Second, the minimum sealing force for graphite packing is 1.0 to 1.25 times the system pressure times the packing area. Third, elongated gland studs are needed to provide axial clearance for the stacked springs. Fourth, the use of disc spring guides prevents hang-up on the stud threads. The preferred location of these guides is internal to the stack and installed over the gland stud. The advantages of the internal location are larger OD for the springs, thus minimizing the required height of the spring stack.

Designers can incorporate into the design of large valves or high maintenance valves a means for hydraulic packing removal to simplify maintenance. This would be most appropriate for control valves, which experience the most severe sealing service since their stems are continuously modulating and have a short seal and/or stem life.

9.3 Pump Seals

The shaft seal of centrifugal pumps should be designed with curved back U-cup ethylene propylene elastomers, since these seals have minimal leakage during off-normal events.

Canned motor pumps may have one external seal, which would be a flange seal for the pump housing. This seal, if present, should incorporate the guidelines given under flange seals. The greatest danger for a canned pump is catastrophic failure. To prevent this, the corrosion behavior of the bearings in large pumps should be examined under conditions representative of the chemistry of the heavy water environment.

9.4 Manufacturing Requirements

Two factors are important in manufacturing, the consistent availability of a qualified component and the consistent production of that component to specification. The unavailability of specified formulations of elastomers has been discussed. For valves, there is little

documentation on their manufacture within many companies, i.e., the tolerances needed for a given leak rate or the loading stress required as a function of the packing material properties. The Canadians found that they required much tighter manufacturing tolerances than most companies could supply.

Our manufacturing guidelines are the following. First, certificates of conformance or compliance would be advisable to ensure that the required material is supplied. These certificates would include the manufacturing history, especially important for bolts, and would document the exclusion of deleterious materials (chloride or sulfur containing). Second, the effects of post-machining stress relaxation or the influence of other structural components need to be considered in a flange design to achieve a seal with a low leak rate. Surface waviness and changes in tolerances increase with physical size, and distortions can occur due to the weight-carrying capacity of a component like a flange.

9.5 Assembly Requirements

Special tools may be required for assembly or disassembly of flanges, valves, and/or pumps. These tools need to be specified and designed if so required. For example, an ultrasonic extensometer should be specified for the assembly of bolted flanges.

9.6 Maintenance Requirements

Achieving seals that are reliable and maintainable requires considerable effort, including the collection of failure rate data for each component and the determination of failure modes, corrective maintenance times, and preventive maintenance times.¹⁴⁰ Design decisions based on analysis of failure modes can improve reliability and maintainability. Physical changes in equipment, systems, or procedures can improve reliability, as can clearly stated specifications for the equipment and factual evidence that the specifications are met.

The failure rate for seals depends on the seal's environment and human action. Failures due to human action can result from improper procedures during field construction. For example, during assembly, flange gaskets which are not properly restrained in grooves can be pinched, thus leaving gaps in the seal. Thus, contract specifications for components should contain clauses on assembly procedures, repair times, and repair procedures. Repair times can be substantially reduced by designers if they provide easy access to components, specify correct maintenance procedures, and specify special assembly/disassembly tools.

Maintenance should include the compilation of maintenance records on each component throughout its lifetime. For example, plant data sheets should be maintained for valves. This data sheet would include the number and type of packing rings; the size of the packing rings; the length of a stuffing box spacer, if needed; the number, size, and stacking arrangement of the disc springs; the maximum and minimum sealing loads; the amount of in-service consolidation accommodated; and the uncompressed spring stack height.

10. RECOMMENDED RESEARCH AND DEVELOPMENT

10.1 Material Tests

A rigorous characterization of the properties of spiral wound graphite filled gaskets, elastomers, expanded-graphite packing materials, bolts, and springs is needed since these properties serve as a basis for designer's calculations.

For spiral-wound gaskets, the leak behavior has to be measured using the technique developed under the sponsorship of BHRA. The properties of the elastomer, EPDM, have to be well characterized if it is chosen as the material of choice to achieve flexible, dynamic seals for safety and control rods. The response to load of graphite packing and Belleville springs, as a function of time, needs to be determined so that the lifetime of these components can be estimated.

10.2 Quality Control Tests

A quality control program is needed for the design, manufacture, installation, and assembly of all coolant component seals and/or joints.¹⁴¹ First, the needed components must be defined. Then, specifications must be set for the components to ensure that they achieve the minimal leak rate. Last, the components have to be qualified in defined tests to ensure acceptable long-term leakage performance. In addition, procedures for the installation and assembly of these components within containment have to be documented.

Four types of tests may be necessary for quality control: qualification tests, validation tests, compliance tests, and development tests. Qualification tests would be done on a prototype component and on a series of identical components under the maximum allowable conditions, regardless of actual service conditions. Validation tests would be done on components that would be subjected to special service conditions. Compliance tests on series-produced components would be carried out under the maximum service conditions observed by all identical components in the plant. These tests would answer two questions: Is the performance in conformity with that of the initially qualified component? Is this component identical to the qualified component as to geometry and its capability to perform the specified functions? Development tests would be done in collaboration with a manufacturer.

Detailed safety analyses may require the demonstration of both operability and leak-tightness under various load conditions that occur in accidents, earthquakes, or high-energy pipe breaks. The condition of equipment in the post-accident environment would be important for future operability of the reactor.

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