INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE PRESSURIZED WATER REACTOR NUCLEAR PLANTS

June 1960

Ebasco Services Inc.
New York, New York
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INVESTIGATION OF TYPES OF SEALS
FOR MAIN COOLANT PUMPS FOR LARGE
PRESSURIZED WATER REACTOR NUCLEAR PLANTS

For
United States Atomic Energy Commission
Under
Contract No. AT(30-1)-2547
New York Operations Office

June 1960

Ebasco Services Inc.
New York, New York
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- Inquiry Specification Sample
- Comparison of Pump Costs

### BIBLIOGRAPHY
I - INTRODUCTION

A - AUTHORIZATION

This report sets forth the results of a study of canned-motor and controlled-leakage pumps for large pressurized water reactor nuclear plants. The report is in accordance with Letter Contract AT(30-1)-2547 dated May 2, 1960 between the United States Atomic Energy Commission and Ebasco Services Incorporated of New York.

B - OBJECTIVE

In general, the objective of this study is to determine the comparative reliability and installed and operating costs of canned-motor and controlled-leakage pump designs for application in a pressurized water reactor installation today, and projected for 1968 operation; including investigation into the scope and adequacy of current research and development programs in this area.

C - CONDITIONS

The following are the principal conditions prescribed by the United States Atomic Energy Commission which established the ground rules for the study:

1) The pump designs studied are to be applicable for service in primary coolant loops of large pressurized water reactors.

2) The pumps are to have the following design characteristics:

   Flow Range 30,000-40,000 gpm
   System Operating Pressure 1,500-2,500 psig
   Total Dynamic Head 40-150 ft
   Speed 1,800 rpm

3) Pump designs and installation and operating costs based on current technology are to be compared to designs and costs projected for placement of pump order in 1965 for plant operation in 1968.

4) A seal research and development program is to be recommended provided current and projected technology to 1965 is not capable of achieving a reliable controlled-leakage pump having the aforementioned characteristics.
FIGURE I - TYPICAL CANNED MOTOR PUMP AS MANUFACTURED BY WESTINGHOUSE ELECTRIC CORP.
FIGURE 2 - TYPICAL CONTROLLED LEAKAGE PUMP AS MANUFACTURED BY INGERSOLL-RAND CO.
D - SCOPE

This report includes a presentation of:

1) Manufacturers' capability to produce pumps of the desired characteristics now and in 1965.

2) Initial and operating costs of both pump types now and in 1965 based upon constant dollars.

3) Manufacturers' seal research and development programs now, planned to 1965, and suggested seal research and development program augmentation to 1965.

4) Estimated pump cost reductions possible in 1965 as a result of a separate controlled leakage pump research and development program.

5) Recommended research and development programs, including a schedule and estimate of costs.

E - PROCEDURE

This report is based on data accumulated by consultation with manufacturers either at their facilities or at Ebasco headquarters. The general procedure was to send manufacturers, in advance, a copy of Inquiry Specification as included in Appendix VII. Most manufacturers completed and returned the specification with such supplementary data as required so as to present a picture of what they were manufacturing as regards canned motor and controlled-leakage pumps, their programs and test facilities, presently available and planned for future. The following outlines the general procedure followed:

<table>
<thead>
<tr>
<th>Companies Examined</th>
<th>Place of Meeting</th>
<th>Inquiry Specification Returned With Requested Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Westinghouse Electric Co.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>General Electric Co.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Allis-Chalmers Mfg. Co.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Byron-Jackson</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Worthington Pump Co.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Ingersoll-Rand Co.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Peerless Pump Co.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bingham Pumps</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>C. H. Wheeler</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Pacific Pumps</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>United Centrifugal Pumps</td>
<td>X</td>
<td>Not Required</td>
</tr>
<tr>
<td>Crane Co.</td>
<td>X</td>
<td>Not Required</td>
</tr>
<tr>
<td>Borg Warner</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>N, R, B.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

This work was performed during the period May 2 to June 30, 1960.
II - DEFINITIONS OF TERMS

Zero-Leakage Pump - A pump used in nuclear field where pump and motor are designed together to form an hermetically sealed unit.

Canned-Motor Pump - An integral motor-pump arrangement where a cylindrical "can" encloses the rotor and the stator. The part of the can in the "air gap" is called the liner.

Wet-Motor Pump - A motor-pump combination where pumped fluid is in direct contact with and flows around motor windings.

Controlled-Leakage Pump - A pump unit where one or both shaft ends penetrate the pressure case of the pump and shaft seals are employed to limit outboard leakage.

Packing - A resilient material so arranged as to follow closely shaft runout and other shaft motions in order to limit leakage.

Face-Type Mechanical Seal - Where two non-resilient materials are arranged that one presses against the other so that necessary clearance to limit leakage can be maintained.

Natural-Circulation Boiler - A boiler in which internal water circulation is maintained by thermal differences and mechanical arrangement.

Forced-Circulation Boiler - A boiler in which internal water circulation is maintained by circulating pumps.

Deaerating Heater - A feedwater heater where steam and water are intimately mixed at saturation temperature for removing entrained gases.

Closed Feedwater Heater - A heater where feedwater is enclosed in tubes and extraction steam surrounds the tubes.

Labyrinth - A series of tortuous passages formed by mechanical means, used to reduce flow.

Fossil Fuel - Fuel such as coal, gas, and oil.

Nuclear Fuel - Fuel such as uranium.

NPSH - Abbreviation for Net Positive Suction Head. This is water pressure required on suction of a pump over and above its saturation pressure to prevent flashing in pump impeller.
**Graphitar** - A bearing material made from carbon and graphite with suitable binding material.

**Stuffing Box** - A space at ends of pump shaft where it penetrates the pump casing that contains a mechanical device for restricting pumped fluid from emergence from casing.
Our principal findings, conclusions and recommendations are:

1) Reliable canned-motor pumps of 40,000-gpm capacity can be built today at an approximate cost of $525,000 each, on multiple-unit orders (1960 costs).

2) Controlled-leakage pumps of similar capacity and reliability are not available today and may not be available in 1965 for 1968 PWR plant operation because pump and seal manufacturers' present and planned seal research programs appear inadequate.

3) Pump and seal manufacturers' seal research and development programs appear inadequate because of limited research appropriations and lack of significant sales potential in this field.

4) A controlled-leakage pump of 40,000-gpm capacity and with a suitable seal design might cost approximately $300,000 (motor included) on multiple orders (1960 costs).

5) Thus, the initial cost incentive to develop a successful pump mechanical seal is in the order of $225,000 per unit. Operating costs further favor the mechanically sealed pump which has an over-all efficiency of about 75% compared to the canned pump efficiency of about 65%.

6) Consequently, an intensive seal research and development program should be initiated to accelerate the realization of a suitable pump shaft seal design.

7) Accordingly, a 4- to 5-year mechanical seal development program is recommended, as outlined in Section VI. The primary objective is that of a shaft seal without external sources of injection water, yet with controlled outboard leakage in the order of 1 gph.

8) The minimum suggested budget for the first 2 years of this program is $700,000. Costs beyond this period are largely
dependent upon the results during the first 2 years and the availability of hot test loops in existing plants. Accordingly, these costs are not estimated herein.

9) The services of a consulting engineer to provide guidance and management of the over-all program is recommended. This will permit exploration of all attractive designs presently being investigated by individual manufacturers as well as efficient execution and integration of all phases of the work.

Furthermore, we find that:

10) Other cost reductions for either pump type are possible with research and development on alternate materials and on relaxation of minimum material test standards for major pump components (such as the casing). Such reductions do not alter the over-all incentive for a controlled leakage design.

11) Thus the desirability of a separate research and development program investigating alternate materials and relaxation of material test standards should be considered.

12) Initial and operating cost reductions with the canned-motor pump design also appear possible. Definition of the technological areas in which such reductions might be made should be the subject of a separate study. As a minimum, alternate can materials and the elimination of several of the concentric pressure cans could be considered.

13) Thus the desirability of a canned pump research and development program should be considered. However, in view of the potential of the controlled-leakage design, such a parallel research and development program may only be justified in terms of military application or in the event the mechanical seal program yields unsatisfactory results.

14) Determination of the over-all objectives, time schedule, cost, etc., of research and development for pump materials, material test standards, and canned-motor pump components cannot be made until further study.
IV - ANALYSIS

A - GENERAL OPERATIONAL FACTORS AFFECTING PUMP SELECTIONS

1 - Comparison of Pumps for Large Nuclear Pressurized Water and Fossil-Fuel-Fired Plants

Reactor primary loop circulating pumps have similar characteristics to boiler circulating pumps used in Combustion Engineering Inc. forced-circulation-type boilers. Operating pressures, temperatures, and dynamic heads are approximately at the same levels, being above 2000 psig, above 600 F, and having total dynamic heads of about 150 ft. The most significant difference is in the required system capacity since boiler circulation pumps circulate boiler water at a ratio of approximately four times the quantity of steam produced, whereas reactor circulating pumps circulate primary coolant at a corresponding ratio of approximately 15 to 1. Since fossil-fired boiler plants produce superheated steam of higher energy content than the saturated steam produced by pressurized water reactor plants, the disparity in pump capacities for equal electric output is actually greater. Reactor circulating pumps require capacities about five times that of boiler circulating pumps.

Boiler circulation pumps are designed with controlled-leakage seals provided with high-pressure water injection ranging from 25-60 gpm per pump. Injection water in-leakage to the cycle is approximately 20% of the total injection water, the balance being discharged to a lower pressure point in the feedwater cycle, usually a deaerater, or a closed feedwater heater. Any small quantities of water leaking to atmosphere presents a plant housekeeping problem although a gross failure of the seal could be a personnel hazard.

Reactor circulating pumps built to date are canned type, zero leakage designs. There is no leakage of radioactive water to atmosphere to present a biological hazard to personnel. Cooling water is provided only to remove motor heat, since neither the windings nor the water lubricated bearings can operate safely above 200 F.
Circulating pumps of the controlled-leakage design proposed for nuclear service have various types and arrangements of shaft sealing devices, provided with high-pressure injection water. Leakage rates are claimed to be held to quantities ranging from drops to several gpm. The design, reliability, and predictability of pump shaft seals is the prime subject of this report since the biological hazards created by leakage to atmosphere are severe.

Circulating pumps, both for fossil-fuel-fired and nuclear plants, are vertical types with the pump casing suction and discharge connections welded directly into the pipelines which support the pumps. As a result the pumps "float" on the pipelines thus eliminating any necessity to absorb pipe movements with flexible connectors which would be required for a fixed base horizontal pump design. Typical pipe movements from hot to cold are 3 to 4 in. for forced circulation boilers and 2 to 3 in. for reactor plants.

A secondary advantage of the vertical pump design for reactor plants is that the horizontal space requirements are considerably less than for horizontal pumps. The circulating pumps are located in the lower portion of the containment vessel where horizontal space is limited, if the containment vessel is a sphere.

Another advantage of vertical pump design is that only one stuffing box is required where pump shaft penetrates casing to couple with motor.

Pressurized water reactor plants require steam generator feed pumps which are generally similar in service and operating characteristics to fossil-fuel-fired plant boiler feed pumps. These pumps require relatively moderate capacities but high total dynamic and discharge heads. Pumps for this service are not within the scope of this report since the steam generator feed pumps operate in the secondary cycle and the possibility of radioactive water leakage from its shaft seals is remote and therefore does not present a major biological hazard.
The following tabulation of pump characteristics illustrates the approximate service requirements of the aforementioned pumps based upon an assumed 250 mw (e) utility type plant.

<table>
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<tr>
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<tr>
<td></td>
<td>Fossil-Fuel- Nuclear Plant</td>
<td>Fossil-Fuel-Fired Plant</td>
</tr>
<tr>
<td>Capacity (total), gpm</td>
<td>128,000</td>
<td>28,000</td>
</tr>
<tr>
<td>Capacity (each), gpm</td>
<td>32,000</td>
<td>7,000</td>
</tr>
<tr>
<td>Suction Pressure, psig</td>
<td>1,950</td>
<td>2,600</td>
</tr>
<tr>
<td>Suction Temp., F</td>
<td>580</td>
<td>674</td>
</tr>
<tr>
<td>TDH, ft</td>
<td>150</td>
<td>125</td>
</tr>
<tr>
<td>Discharge Pressure, psig</td>
<td>1,995</td>
<td>2,630</td>
</tr>
<tr>
<td>BHP (each)</td>
<td>1,150</td>
<td>175</td>
</tr>
<tr>
<td>BHP (total)</td>
<td>4,600</td>
<td>700</td>
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</table>

2 - Operational Comparison of Reactor Circulating and Boiler Circulation Pumps

Forced circulation boilers are designed primarily to reduce the height required of a natural circulation fossil-fuel-fired boiler, and to improve the flow patterns and heat transfer in the water wall tubes. Nevertheless, the water surface in the boiler drum is about 80 ft above the inlet of the boiler circulation pump. Since the boiler drum water is mixed with cooler incoming feedwater before flowing to the boiler circulation pump, the available net positive suction head of 100-300 psi is created by overpressure as well as by static height.

The static height available in a PWR nuclear plant is generally not more than 40 ft. This is a function of the containment vessel size which is kept to an economic minimum compatible with equipment, shielding, and free volume requirements. At economic pump speeds
the net positive suction head provided by static height alone is not sufficient to prevent the reactor water from flashing into steam and causing pump cavitation. However, for the pressurized water concept the reactor system pressure is several hundred psi above that corresponding to the reactor water temperature to prevent boiling. The reactor over-pressure system maintains the system pressure safely above the reactor saturation pressure by the controlled boiling or condensing of steam in a separate pressurizer.

The NPSH available due to static and overpressure system is generally sufficient for economic pump selection, although during start-up and transients, full NPSH may not be available.

The pressurized water reactor plant coolant cycle is best described as a closed loop load to a centrifugal pump. Figure 3 illustrates such a closed loop and shows the main coolant pump discharging high-pressure water through the reactor core where it is heated. The hot coolant passes through the tube side of the steam generator where heat is transferred to the feedwater on the steam generator shell side thus generating steam, but at a lower pressure. The cooled water returns to the suction of the coolant pump to repeat the cycle.

Reasonable pump and steam generator sizes require that a large pressurized water reactor plant have a number of parallel closed loops. Present reactor plant design incorporate as many as four parallel loops. Regardless of the number of loops designed into the plant, each pump operates on its individual circuit. In certain instances either two-speed motors, or a variable frequency power supply, is furnished to provide flow regulation. The reactor, however, is a common water channel for all loops, and the resistance of the reactor core as a flow channel depends upon the quantity of flow through it, which is a function of the number of pumps in operation. Any change in reactor core pressure loss, causes each pump to reposition itself on its capacity-head curve, thus varying the resistance through each pump's steam generator, piping, valves, etc. The coolant pumps must therefore be designed to operate satisfactorily over the expected flow range, since no regulating device exists in the loop and all loop elements have fixed resistance at a given rate of flow and the loop flow quantity remains essentially constant.
PWR PRIMARY SYSTEM

STEAM
FEEDWATER
FEEDWATER
STEAM
TO TURBINE
FROM CONDENSER
FROM CONDENSER
TO TURBINE

AUXILIARY EQUIPMENT

PRESSURIZER

FIGURE 3 - TYPICAL MAIN COOLANT CLOSED LOOP ON PRESSURIZED WATER REACTOR PLANT
B - DESCRIPTION OF ZERO LEAKAGE AND CONTROLLED-LEAKAGE PUMP TYPES

Zero leakage pumps are designed with a pressure-type casing completely enclosing both the motor and the pump. No portion of the rotating elements penetrates the pressure tight casing to afford a path for coolant leakage and are therefore designated as "zero leakage" pumps. Several small stationary elements of the pump penetrate the casing and are welded or otherwise sealed to prevent fluid escape to atmosphere. These include the motor leads and cooling water tubing. These elements are readily sealed with proper design and manufacturing techniques and do not present any major leakage problem. The motor casing is bolted to the pump casing using either all metal gaskets, or combination metal-asbestos gaskets. For final assurance against leakage at this joint a seal weld is often provided. The motor head is a similar bolted joint. In the canned-motor pump, to maintain dry windings, the stator and rotor are completely enclosed with thin sheets of metal such as stainless steel or Inconel. These Liners, which resemble a can, have resulted in the pump being termed a "Canned-Motor" pump.

A small quantity of coolant water fills the space between the stator and rotor liners. This water is circulated by an auxiliary impeller to serve as lubricant for the water lubricated carbon-graphite journal and thrust bearings and to carry away motor heat. The water flows out of the pump casing through a spiral tube coil wrapped around the motor casing to the top of the motor casing to complete the circuit. An external pressure tight casing covers the spiral coil as added assurance against leakage and to provide a flow channel for an external cooling water system to remove heat.

All materials in contact with the coolant, except for the water-lubricated bearing surfaces, are stainless steel or Inconel since they are in direct contact with the fluid being pumped. The over-all efficiency of a canned-motor pump may be 10-12% less than a controlled-leakage type pump or about 65-69%. Since the two Inconel cans which
seal the rotor and stator fill the air gap, electrical efficiency is low; and since the rotor turns in the water filling the can clearance space, hydraulic losses are high. About 50% of the motor loss is attributed to the cans and 35% to the hydraulic losses. The balance of the extra losses are a result of the relatively long and narrow motor which is required to minimize the wall thickness of the pressure tight casing. The hydraulic end of the pump is of equal efficiency to any other similar pump type.

Several other types of zero leakage pumps have been devised to satisfy the requirements of various reactor concepts and other nuclear and chemical applications. Current designs for water pumping include the aforementioned canned rotor and stator type, the submerged or wet winding type, the gas filled motor, and the oil filled motor. Current designs of pressurized water reactor plants have incorporated only the canned-motor type, but the wet winding could hold promise due to its successful use on boiler circulating service. Although promising in smaller size low-pressure applications, wet-motor pumps have not been used on PWR systems to date; accordingly they are not reviewed in this report, and are only mentioned as varieties of zero leakage designs.

A vertical controlled-leakage-type pump has an overhung impeller and therefore only one casing penetration is required for the shaft. The shaft seal type and arrangement determines the total quantity of injection water required, if required at all. A commercial type of vertical motor is coupled to the hydraulic end. The pump casing, impeller, and shaft are presently made of stainless steel when used for nuclear service. The motor and coupling however can be built of the less expensive carbon steels. Motor efficiencies are the maximum available in current technology, ranging from 92-95%. Thus the over-all pump efficiency is at its possible maximum except for efficiency losses attributable to seal friction and injection water pumping requirements. Over-all efficiencies may range from 75-81% compared with 65-69% on canned-motor units.
Conventional boiler feed pumps have one great advantage, where limited leakage pump designs are utilized, that cannot be designed into pumps for main coolant service. When a high-pressure boiler feed pump is designed the pump designer arranges the pump impellers so that they are opposed, in order to reduce thrust as much as possible. In so doing, the high-pressure impellers are arranged in the center of the pump with the suction impellers or low head stages outboard. This has the advantage of low-pressure, low temperature fluid being placed on the glands where the pump shaft penetrates the casing. On high-pressure boiler feed pumps today where discharge pressures of 2500 to 3000 psi are encountered, the actual pressure on pump glands can be held to relatively low pressure in neighborhood of 150 to 400 psi. Temperatures in this case are not over 400 F since condensate has not gone through its final HP heater stages of heating.
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<tr>
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<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Location</td>
<td>Shippingport, Pa.</td>
<td>Italy</td>
<td>Aircraft Carrier</td>
<td>Rowe, Mass.</td>
</tr>
<tr>
<td>Manufacturer</td>
<td>Westinghouse</td>
<td>Westinghouse</td>
<td>Westinghouse and General Electric</td>
<td>Westinghouse</td>
</tr>
<tr>
<td>Capacity, gpm</td>
<td>18,300</td>
<td>24,500</td>
<td>11,800</td>
<td>23,700</td>
</tr>
<tr>
<td>TDH, ft</td>
<td>340</td>
<td>170</td>
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C - DESCRIPTION OF CONTROLLED-LEAKAGE SEALS

Conventional thermal power stations for many years have been faced with the problem of proper centrifugal pump stuffing box designs. As pressures and temperatures of pumped fluids increased, the various designs were altered in order to maintain reliability and relatively low maintenance.

When the pressurized water reactor station became a reality, stuffing box designers who had previously been working with moderate temperatures and pressures of 450°F, 350 psig, were immediately faced with pressures of 2000 psig and temperatures of 600°F. This placed a very serious block in the way of the more gradual development as experienced in thermal station design. Since up to that time little research had been done and the developed seals were inadequate for the purpose, the canned-motor pump was developed to the design criterion of "zero leakage."

At present stuffing box designs have progressed to a great degree although as yet no type of seal has been developed where a definite prediction can be made on the amount of seal leakage or a definite value placed on its life expectancy or reliability.

The limited leakage designs, however, have great possibility of being developed where they could be used with high reliability on pressurized water type nuclear station service.

There are many factors that affect stuffing box design, and the life of packing or seal as the case may be. These are:

1) Pressure
2) Temperature
3) Speed
4) Type of Packing or Seal
5) Cleanliness of Fluid Handled
6) Materials
7) Packing Skill and Experience
8) Machinery and Assembly

Any type of pump seal is a pressure breakdown device. Since it is a breakdown device it must absorb energy resulting in the generation of heat which has to be removed by either cold water jacketing, leakage
of water through the packing or by conduction through surrounding metal to atmosphere. For large pressurized water nuclear plant main coolant service, the design must be such that seal materials are capable of withstanding the high temperatures and pressures encountered, be inert to the highly corrosive water being pumped throughout the closed system, and be of such design that if a failure of the seal occurs, the result will be predictable so far as the out-leakage from the system is concerned.

Following is a summary of various controlled-leakage seals that are available today illustrating the various designs, their limitations, and what can be expected from them.

1) Packed Stuffing Box
2) Labyrinth Seal
3) Pressure Breakdown Bushing
   a - Fixed
   b - Floating
   c - Ring Seal
4) Face-Type Mechanical Seal
5) Elastomeric Seal
6) Combination Seal

1 - Packed Stuffing Box

The packed stuffing box illustrated in Figure 4 is a common type which is widely used on centrifugal pump service today. This consists of a shaft sleeve usually of 12 to 14% chrome stainless steel hardened to about 500 Bineil hardness or hard surfaced with stellite. This shaft sleeve is usually pressed on to the pump shaft and held in place by means of a shaft nut. Individual rings of packing are then formed over the shaft sleeve and inserted inside the stuffing box and held in place by a packing ring which is tightened by means of bolts forcing the packing against the sleeve and the sides of the stuffing box. The packing itself is made from organic or inorganic materials depending on the pressure and temperature conditions and the type of fluid being pumped.

This type of packing is usually recommended for service not over 150 psig and 350°F but there are units in service with pressures of 300 psig and 400°F. Where pump fluids are encountered over 250°F, a
FIGURE 4 - PACKED STUFFING BOX
water cooling jacket and a quench gland as outlined in the illustration has to be provided. This type of packing has a high maintenance factor and skilled personnel are necessary for its installation and maintenance, for with this type of packing its life expectancy and success depends entirely on the operator who installs and maintains it. It is subject to manual adjustment over its life and with the best of care and supervision usually has a life expectancy of not over one year. This type of packing does not appear to have application in pressurized water main coolant pump service when utilized by itself but may be applicable in some instance as a backup on a combination type seal.

2 - Labyrinth Seal

The labyrinth seal falls in the category of a clearance seal. It is a device to limit leakage between a rotating shaft and a stationary housing by maintaining a closely controlled annular clearance between the two. The labyrinth has the advantage of high reliability, simplicity and flexibility in the selection of materials. It consists essentially of one or more thin strips which are attached either to the housing or to the shaft itself. Design clearances between the strips and the housing or shaft sleeve are maintained at low closely controlled values. This is illustrated in Figure 5.

In a seal labyrinth of the straight-through type as in Figure 5a, sleeve effectiveness is limited by a carry-over of energy associated with each stage into the succeeding stage. In order to reduce the carry-over a refinement of the simple labyrinth can be introduced as shown in Figure 5b where the addition of grooves or steps between throttlings serves to deflect the expanding fluid into the space between the strips which space tends to act as a stilling chamber. This refinement, however, has not found too much application in pump design since it increases manufacturing costs and introduces an assembly and alignment problem.

The straight-through labyrinth as shown in Figure 5a while not applicable to main-cooling pumps for pressurized water reactor service by itself, does have application when used in conjunction with other
LABYRINTH SEAL

FIGURE 5a - STRAIGHT-THROUGH TYPE

FIGURE 5b - STEPPED TYPE
types of seals. The high-pressure differentials encountered and the relatively large clearances required for the labyrinth would cause leakages far in excess of that permitted for pressurized water service.

The leakage of any labyrinth is a function of the clearance area which is directly proportional to radial clearance. Inasmuch as any rotating shaft undergoes radial motions resulting from an accumulation of bearing dynamic deflections and thermal distortions there is a practical minimum for labyrinth clearance to avoid rubbing which could cause strip burning. Any rub while not necessarily destroying the labyrinth strip will cause increased radial clearance and consequent higher leakage. Labyrinth clearances normally are in the neighborhood of 0.010 to 0.020 in. An interesting and effective variation of the labyrinth is a device called a wind back. This consists of the stationary sleeve surrounding the shaft on the downstream side of the seal. The sleeve is threaded opposite to that of shaft rotation so that centrifugal force throws seal leakage in contact with the shaft radially outward forcing the liquid to wind back through the threads to a leak-off tap. This device while only usable for zero pressure differential could have application on main-cooling pump seals in conjunction with other arrangements.

3 - Pressure Breakdown Bushing

The pressure breakdown bushing is a clearance seal consisting essentially of a close fitting stationary sleeve within which the shaft rotates. Leakage from the high-pressure end of the bushing to the low-pressure end of the bushing is controlled by the restricted clearance between shaft and bushing. Ideally, the bushing and shaft are perfectly concentric and no rubbing takes place. These bushings themselves fall into two categories consisting of fixed and floating bushings. The principal advantage offered by both the fixed and floating bushings over labyrinth is that of considerably lower leakage per unit length.

a - Fixed

The fixed bushing seal presents advantages and disadvantages similar to those found with the simple labyrinth. Generally, the seal consists of a long sleeve fixed to the machine housing within
which the shaft rotates with a small clearance. It requires, however, fairly large clearance, the same as the labyrinth, due to bearing tolerances, dynamic deflections and thermal distortions. For the larger clearances, longer assemblies are required to keep leakage within reasonable limits. These longer bushing assemblies, however, tend to further aggravate alignment problems.

The fixed bushing by itself is not applicable to main-coolant pumps. However, it has possibilities when used in conjunction with other sealing arrangements.

b - Floating

Floating bushings are clearance seals which are free to move in a radial direction relative to the shaft and to the machine housing. These are illustrated in Figure 6. They offer the considerable advantage that very close annular clearances can be used. Radial clearances are generally .0015 in. per inch of shaft diameter. Their floating characteristic permits them to move relatively freely with shaft motions and deflections thereby avoiding the effects of rubbing. The only constraints to radial motion of the seal ring are the inertia of the ring, friction between the end of the ring and the chamber wall against which it is seated by hydraulic forces and any springing which may be used.

As a rotating shaft tends to move in a position eccentric with the bushing, pressure in excess of the applied pressure is built up in the constricted portion of the annulus. As soon as the net radial forces corresponding to the new pressure distribution exceeds the force required to overcome inertia and friction, the bushing moves radially until equilibrium is once again established.

Two distinct varieties of floating bushings exist. The first of these which is the single floating bushing as shown in Figure 6a. An axial spring is used to maintain a seal at the radial wall of the chamber under the low-pressure differentials. Where high differentials are encountered the spring may be omitted.

Where extremely high-pressure differentials are encountered it is bad practice to rely on a single bushing to take the entire pressure breakdown.
c - Ring Seal

The second variety is termed a floating ring seal as illustrated in Figure 6b. It is schematically exactly like the single floating bushing with the exception that a multiplicity of bushings or rings are used. Each ring is independent of the others in its motion and this independence, combined with the short length of each ring permits considerable skew between shaft and housing with little sacrifice in sealing effectiveness. Assembly length is increased to the degree that diaphragms are required to separate the rings but the percentage increase in length is not too great.

In fixed bushing seals the possibility of rubbing must always be considered. Materials for these seals must be such that any rub will generally be localized and not affect the bushing as a unit. With the high temperatures and high pressures encountered in main-coolant pump service, bronze and aluminum alloys are generally ruled out. Floating bushing and floating seal rings being relatively unconstrained as to radial movement permit the designer a much wider choice of materials than do fixed bushings. Shaft scoring is minimized even with clearances in the order of a few thousandths of an inch, per inch of shaft diameter and therefore hard materials may be used if necessary. For high-pressure water service the combination of a stellite floating ring with chrome plated or stainless steel shaft or shaft sleeve may prove quite effective, since the slightly higher coefficient of stellite causes growth away from the shaft or sleeve material in the event of a rub.

In connection with sleeve rub, water, such as is used in primary cooled service, if permitted to leak through a seal would reach a saturation pressure and flash into vapor. It is necessary to take into account the effect of any entrained solids in the sealed fluid. These will be deposited as "crud" within the seal at the point where flashing takes place, and can build up to a point where the seal becomes inoperative. It is therefore necessary to provide a cooling fluid at a pressure slightly higher than the system pressure. The sealed medium then flows inward into the machine and out through the seal.
FLOATING BUSHING OR RING SEAL

FIGURE 6a - SINGLE FLOATING BUSHING

FIGURE 6b - MULTIPLE FLOATING BUSHING
4 - Face-Type Mechanical Seal

The mechanical seal, as it is often called, is a face-type contact seal which is coming more into use on conventional pump designs. The mechanical seal is at present being used for applications requiring a high order of seal effectiveness where relative axial motion between shafting and casing is limited. Figure 7 illustrates a typical mechanical face seal wherein a seal ring keyed by a rotation lock to the stationary housing contacts a shoulder or mating ring that is constrained to rotate with the shaft. Primary sealing takes place in the radial plane of motion between the shoulder and the projecting dam on the seal ring face. This type of seal shows greatest potential for main-coolant pump service as discussed in Section IV E.

5 - Elastomeric Seal

Elastomeric seal packings in the form of O rings, quadrings, vee rings, and many other types of flexible members constitute the class of secondary seals in widest use today. These seals make use of rubber, teflon, nitron A, and large numbers of other materials which are used in a great array of sizes. These seals are not in themselves applicable to main-coolant pump service but are necessary to seal stationary surfaces of the mechanical seal assembly. Several types are shown in Figure 8.

6 - Combination Seal

Combination Seals in use today take many and varied forms. Those used for boiler feed service in conventional plants are illustrated in Figure 9 Gland Suction Bleed-Off and Figure 10 Condensate Bleed System. It will be noticed in Figure 9 for the suction bleed-off that a breakdown type of bushing is utilized with a bleed-off to a lower-pressure heater backed up by a labyrinth-type ring and conventional packing with a snuffing connection. This type of combination seal is not applicable to main-coolant pump service due to the high leak-off encountered. Since the fluid escaping from the breakdown bushing is piped to a lower-pressure heater, the conventional packing can be held to pressures in the neighborhood from 75 to 150 psi. The condensate bleed-in system
FIGURE 7 - FACE-TYPE MECHANICAL SEAL
FIGURE 8 - ELASTOMETRIC SEALS
FIGURE 9 - COMBINATION SEAL FOR BOILER FEED PUMP SUCTION
FIGURE 10 - COMBINATION SEAL FOR BOILER FEED PUMP SUCTION
as illustrated in Figure 10 is also used for boiler feed pump service in conventional station design. It will be noted that high-pressure condensate is bled into the breakdown bushing where some leakage flows into the pump with the remainder piped to a lower-pressure heater. In this type of seal the pumped fluid does not come into contact with the conventional packing. Here again this type of seal is not applicable to main-coolant pump service since the leakages are high. However, the general principle of applying high-pressure water to a breakdown bushing which would be discharged into the pump with some small amount of leakage to a waste system seems to hold good promise for an effective seal required for nuclear service.

Figures 11, 12 and 13 indicate several different arrangements of combination seals available. All of these with further research, development, and testing may be applicable to main-coolant pump service.

D - COMPARISON OF CANNED AND CONTROLLED-LEAKAGE PUMPS WITH REGARD TO:

1 - Cycle Efficiency and Auxiliary Power

Canned-motor pumps have approximately 10% less over-all efficiency than a controlled-leakage pump of the same capacity and head. For a plant of 300 mw gross output the canned pump system would cause a reduction of about 700 kw in net output and a corresponding loss of 0.1% in plant efficiency.

2 - Seal Water Supply System

Canned pumps do not require a high-pressure cooling or lubricating water supply. A small quantity of reactor water circulates in the motor casing and through an integral cooling coil to lubricate the Graphitar bearings and to carry away motor heat. Low-pressure service water from the plant's service water system serves as a heat transfer medium.

Depending upon the seal type and design, controlled-leakage pumps may require high-pressure injection water. The quantities vary from as high as 75 gpm per pump for a pressure breakdown bushing type
FIGURE 11 - COMBINATION SEAL - POSSIBLE ARRANGEMENT APPLICABLE FOR MAIN COOLANT PUMP SERVICE
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**Figure 12** - Combination Seal - Possible Arrangement Applicable for Main Coolant Pump Service
seal to 1.0 gpm per pump for a face-type mechanical seal. One face-type seal arrangement has been designed for zero injection water, but for small shafts and moderate pressures and temperatures.

A pressurized water nuclear plant may not have a source of high pressure injection water, particularly if its control rod drive seals are of the non-injection type. If such is the case, separate means for a high pressure water source must be provided. Presently available equipment for high pressure water service is either a small boiler feed type pump, or a reciprocating pump with pulsation damping auxiliaries. Either method requires a considerable capital expenditure for equipment, instrumentation and controls in the order of $30,000 and introduces another critical system, the failure of which affects continuity of plant operation.

3 - Seal Water Treatment

Canned pumps are zero leakage designs and therefore have no effect on the waste treatment system.

Controlled-leakage pumps have outboard leakages varying from 50 gpm per pump for pressure breakdown bushings to near zero for face-type mechanical seals depending upon the seal type and design. High rates of leakage can affect the design of the cleanup demineralizer system, the make-up demineralizer system or the waste disposal system, depending upon whether leakage recovery is economic or not. For this reason, it must be concluded that a seal design with the minimum outboard leakage will result in an over-all plant economy.

4 - Piping, Instrumentation and Controls

Canned-motor pumps require thermocouples located in its motor windings and in its cooling circuit to indicate that temperatures do not exceed 200 F (above this temperature both the winding insulation and the Graphitar bearings may fail). Normally, the cooling circuit is designed to maintain 150 F in the motor casing.

Canned-motor pumps also require proper venting of the motor casing since gases produced by radiolytic dissociation of reactor water may collect in the upper end of the motor casing. Venting the
casing may be a controlled continuous or intermittent process, but must be properly instrumented to guard against venting failures.

Controlled leakage pumps having seals of the injection and/or leak-off type require pressure controls and piping system to maintain design water flow rates. The differential pressure for injection type seals is usually about 50 psi. Thermocouples located in the seal chamber, or at strategic locations in the seal piping circuit are required to indicate the seal operating temperatures. A rise in temperature at the seal is a strong indication of increased seal friction and potential trouble. The quantity of seal leakage should also be measured, since above normal leakage is also an indication of potential trouble.

5 - Maintenance

Controlled-leakage pumps, regardless of seal design, can be serviced and have minor parts replaced at the job site. Only a major accident involving parts other than the seal would require factory repair.

Canned pumps requiring only guide or thrust bearing repair or replacement can be repaired at the job site. Any accident involving the rotor or stator cans, windings, shaft, cooling system and pressure tight casings will require factory repair.

6 - Motor and Motor Insulation Selection

Controlled-leakage pumps utilize standard motors which are commercially available in a wide range of horsepowers and speeds. Motor insulation may be of standard boiler house quality since there is less exposure to radioactive reactor water. Pump thrusts are readily taken by the motor bearings.

Canned pump motors are specifically designed for each application and are presently not available in a wide range of horsepower and speed. The complex motor design is described elsewhere in this report, but in particular a Kingsbury type of water lubricated thrust bearing with graphite faces and water lubricated graphite guide bearings must be utilized.
Canned-pump motor insulation must be capable of withstanding radioactivity from the adjacent reactor water which circulates in the motor.

7 - Pump and Motor Material Selection

Present pressurized water reactor plant purity standards require that the pump casing, impeller and shaft be of stainless steel material for either the canned or controlled leakage pumps. Controlled-leakage pump motors are not in contact with the main coolant loop water and therefore motor and coupling materials may be carbon steel as normally utilized in commercial motor practice. Canned-pump motors require Inconel cans to seal the windings from the reactor water and stainless steel material for any other motor part in contact with this water.

8 - Operating Flexibility

Both pump types have equal flexibility with regard to normal operation. However, in the event of pump maintenance problems, the controlled leakage pump is more easily repaired in the field. On a motor failure, the canned-motor pump must be returned to the factory for repair, but a spare motor can be easily installed on a controlled-leakage pump.

E - SELECTION OF THE MOST ADVANTAGEOUS SEAL DESIGN AND SYSTEM

Considering the aforementioned factors, the most advantageous seal type is that which most nearly satisfies the following:

1 - Reliability

A minimum service life of about two years. This will permit preventive seal maintenance during the usual annual downtime for general plant maintenance.

2 - Predictability

The seal design should be insensitive to minor variations in manufacture and assembly with regard to leakage and service life.
3 - Minimum Outboard Leakage

The outboard leakage should be kept to a minimum to result in the lowest over-all plant cost and least influence in the design of auxiliary systems.

4 - Minimum Injection Water Supply

The additional plant auxiliary equipment required to supply high-pressure injection water can represent a large investment and maintenance cost. Thus a seal requiring minimum injection water is desired.

The face-type mechanical seal, arranged so as not to require high-pressure injection water and capable of performing with minimal outboard leakage, appears most likely to satisfy these criteria. This seal arrangement is illustrated in Figure 14.
NOTE: WATER MUST BE COOLED TO 160°F THRU HEAT EXCHANGER.

FIGURE 14 - FACE-TYPE MECHANICAL SEAL ARRGT.
NOT REQUIRING INJECTION WATER
A - GENERAL

Canned-motor pumps have been built up to 25,000-gpm sizes with system pressures of 2500 psig and temperatures approaching 600 F. Canned pumps are presently being designed for a capacity of 32,000 gpm for a boiling water reactor plant. Controlled-leakage pumps of similar hydraulic design have approached the same conditions but have not been built to the same capacity. The upper limit is presently 14,000 gpm with pressure and temperature conditions the same as canned-motor units. Designs are being readied for 18,000-gpm and 22,000-gpm units. However, these pumps, designed for forced-circulation boilers, are unacceptable for nuclear service since they require large quantities of high-pressure injection water and have high outboard leakage.

Canned-motor units have been used for some forced-circulation boilers to eliminate the need of high pressure seal injection water during start-up or when the boiler is not steaming but under full pressure. The controlled-leakage pumps under these conditions require either a boiler feed pump to be in operation to supply the injection water required, or alternatively to provide small capacity, high-pressure injection water pumps.

This same condition is in general applicable to main coolant service in a pressurized water nuclear plant. If injection-type, controlled-leakage pumps are employed, a high pressure source of water must be available. The source may be the buffer-seal system if provided for reactor control rods, but of augmented capacity; or separate small capacity, high pressure injection water pumps.

Not all manufacturers investigated in this report produce all varieties of pumps, for instance, canned-motor, controlled-leakage or wet-motor pumps. Most manufacturers do not manufacture seals for controlled-leakage pumps but rely on other manufacturers who specialize in this type of work. Some pump companies, however, are designing seals for special large size shaft applications or are working with seal manufacturers on their development.

The wet-motor pump, which is a zero leakage design, has been referred to in certain parts of this report even though it was not included
specifically in the scope. This was done as there appears to be application of this pump to main coolant service. It was found, however, that much research is required on motor insulation due to possible radiation and particle deposition in the windings. Therefore, wet-motor pumps are mentioned only where it was felt necessary to correlate information between the canned-motor and controlled-leakage designs. The wet-motor pumps greatly increase motor efficiency by elimination of stator and rotor cans.

Table 3 outlines in general the status of each company examined for this report. This is subject to change at any time. In addition to pump companies, Crane Packing Co. and Borg-Warner, seal manufacturers, presented their views on sealing arrangements.

Table 3

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<td>N</td>
<td>N</td>
</tr>
<tr>
<td>Allis-Chalmers Mfg. Co.</td>
<td>P (1)</td>
<td>P</td>
<td>P</td>
<td>(3)</td>
<td>(1)</td>
</tr>
<tr>
<td>Byron-Jackson</td>
<td>P (2)</td>
<td>P (2)</td>
<td>P</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>Worthington Pump Co.</td>
<td>P (2)</td>
<td>N</td>
<td>P</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>Ingersoll-Rand Co.</td>
<td>N</td>
<td>P (2)</td>
<td>P</td>
<td>P</td>
<td>(3)</td>
</tr>
<tr>
<td>Peerless Pump Co.</td>
<td>P (2)</td>
<td>P (2)</td>
<td>P</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>Bingham Pumps</td>
<td>P (2)</td>
<td>P (2)</td>
<td>P</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>C. H. Wheeler</td>
<td>N</td>
<td>N</td>
<td>P</td>
<td>N</td>
<td>N</td>
</tr>
<tr>
<td>Pacific Pumps</td>
<td>N</td>
<td>P (2)</td>
<td>P</td>
<td>P</td>
<td>N</td>
</tr>
<tr>
<td>United Centrifugal Pumps</td>
<td>N</td>
<td>N</td>
<td>P</td>
<td>P</td>
<td>N</td>
</tr>
<tr>
<td>Crane Packing Co.</td>
<td>N</td>
<td>N</td>
<td>N</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>Borg-Warner</td>
<td>N</td>
<td>N</td>
<td>N</td>
<td>P</td>
<td>P</td>
</tr>
</tbody>
</table>

P - Producing
N - Not producing
(1) Under curtailed program
(2) Hydraulic ends only
(3) Indicated work being done of proprietary nature
As indicated, the major pump manufacturers are in a position to build large pump hydraulic ends based on present-day knowledge. However, the sealing system for these pumps has not been designed, tested or proven.

The canned-motor pumps offered at the present time are a tested and proven design and when properly vented and with system cleanliness assured, offer satisfactory reliability.

It has not been possible, during the time allotted, to accumulate any degree of operating experience on the canned-motor or controlled-leakage pumps in civilian or naval service with the exception of those applied to forced-circulation boilers.

Table 4 illustrates merits of controlled-leakage pumps versus canned-motor pumps. During the course of this investigation pump manufacturers made the claim that hydraulic end costs, both for canned-motor and controlled-leakage pumps, are 30-50% greater for nuclear grade castings than for commercial grade castings. Pump casings for nuclear service are stainless steel casings subject to rigorous X-ray, dye penetrant and hydrostatic tests which inevitably result in rejections of many raw castings at the foundry or expensive corrective work by the pump manufacturer. Much of the corrective work is required by test standards originally devised for naval applications with slight relaxations for civilian reactors. Interpretation of test readings lean heavily toward the conservative side because pump operating experience is limited, and there is little correlation of experience to test findings.

B - **COST (1960 LEVELS)**

The high initial cost of canned-motor pumps is a large factor in its disfavor, the motor itself accounting for about 60% to 75% of the initial cost. This high initial cost would appear justified for their application to marine and naval service which are subject to hazardous situations, but should be properly evaluated when used for central station service, due to the competitive nature of the industry and its relatively safer stationary operation.

From discussions with various manufacturers it has been estimated that a controlled-leakage pump of 40,000 gpm with a total dynamic head of 150 ft with a system operating pressure and temperature of 2500 psi at 625 F in stainless steel material would cost approximately $225,000 to $250,000 without motor or development costs. A 2000-hp, dripproof,
<table>
<thead>
<tr>
<th>Controlleed-Leakage Pump</th>
<th>Canned-Motor Pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>G</td>
<td>G</td>
</tr>
<tr>
<td>P</td>
<td>P</td>
</tr>
</tbody>
</table>

G - Good  
F - Favorable  
P - Poor
<table>
<thead>
<tr>
<th>Engineering Office</th>
<th>Westinghouse Electric Atomic Equipment Corp., Department</th>
<th>General Electric Atomic Motor Product Section</th>
<th>Worthington Corporation</th>
<th>Byron-Jackson Pumps, Inc.</th>
<th>Peerless Pumps</th>
<th>Bingham Pumps</th>
<th>Peerless Pumps</th>
<th>Ingersoll-Rand</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump Types Manufactured</td>
<td>canned</td>
<td>canned</td>
<td>controlled leakage</td>
<td>canned and controlled leakage</td>
<td>controlled leakage</td>
<td>controlled leakage</td>
<td>controlled leakage</td>
<td>controlled leakage</td>
</tr>
<tr>
<td>Present Maximum Pump Size, gpm</td>
<td>25,000</td>
<td>25,000</td>
<td>10,000</td>
<td>7,000</td>
<td>7,500</td>
<td>5,000</td>
<td>10,000</td>
<td>14,000- 22,000</td>
</tr>
<tr>
<td>Manufacturing Capability, gpm</td>
<td>as req'd</td>
<td>as req'd</td>
<td>as req'd</td>
<td>as req'd</td>
<td>as req'd</td>
<td>as req'd</td>
<td>as req'd</td>
<td>as req'd</td>
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<tr>
<td>Component Testing</td>
<td>X-Ray</td>
<td>yes</td>
<td>by vendor</td>
<td>by vendor</td>
<td>by vendor</td>
<td>by vendor</td>
<td>by vendor</td>
<td>by vendor</td>
</tr>
<tr>
<td></td>
<td>Dye Penetrant</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td></td>
<td>Helium Leak</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td></td>
<td>Hydrostatic</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Performance Testing</td>
<td>Maximum Flow, gpm, cold</td>
<td>32,000</td>
<td>35,000</td>
<td>110,000</td>
<td>60,000</td>
<td>50,000</td>
<td>40,000</td>
<td>40,000</td>
</tr>
<tr>
<td></td>
<td>Maximum Flow, gpm, hot</td>
<td>32,000</td>
<td>35,000</td>
<td>1,000</td>
<td>7,500</td>
<td>none</td>
<td>7,500</td>
<td>none</td>
</tr>
<tr>
<td></td>
<td>Maximum Temperature, F</td>
<td>600</td>
<td>650</td>
<td>250</td>
<td>650</td>
<td>-</td>
<td>650</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Maximum Pressure, psig</td>
<td>2,500</td>
<td>2,500</td>
<td>1,500</td>
<td>7,500</td>
<td>-</td>
<td>2,500</td>
<td>-</td>
</tr>
<tr>
<td>Seal Manufacturing</td>
<td>no</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Seal Test Laboratory</td>
<td>no</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
</tbody>
</table>
Class A insulated motor would cost about $50,000 thus totaling about $275,000 to $300,000 for the controlled-leakage unit. Subsequent units would not vary too much in price from this since the motor is standard as well as pump and components.

A comparable canned-motor pump would cost approximately $700,000 for the lead unit including design and test, with a reduction to $450,000 for subsequent units. The high initial cost of the lead unit is due to development charges on the motor which must be designed as a new entity. The average cost per pump for a four-loop system is approximately $525,000. The first cost savings with controlled-leakage pumps are substantial.

Operating costs also favor the controlled-leakage pump principally because of its over-all efficiency of 75% compared with 65% for the canned-motor pump. This efficiency difference, evaluated at 40,000 gpm, 600°F and 150 ft TDH, results in 157 kw extra power consumption for the canned-motor pump. At 90% plant factor, 28% cycle thermal efficiency and fuel cost of $0.30 per million Btu, the extra annual fuel cost for one canned-motor pump is $4,525. Capitalized at 12% this is equivalent to $37,700. The associated capability charge at $100/kw is $15,675 per pump. In summation, for a large four-loop system, the differential operating and capability costs, in favor of the controlled-leakage system, is equivalent to an initial cost of $213,500.

The high initial and operating costs of canned-motor pumps illustrates the significant incentive to develop an economic and reliable shaft seal.

C - MANUFACTURERS' RESEARCH AND DEVELOPMENT PROGRAMS

Of the various companies covered in this report some have research and development programs underway, mainly of limited scope, dealing with specific design and operating problems on their standard line of pumps. Some companies consider their research and development programs proprietary while others were willing to generalize but not outline specific details. Listed below is a summary of current programs as reported by each manufacturer. It will be noted that specific level of effort, schedules, etc. are not given mainly because the manufacturers have not projected their programs for any specific purpose or any time limit.
1 - Current Manufacturers' Programs in Seal Research and Development

a - Worthington Corporation

A combination seal development program entitled Project P-915 with test facilities to 3600 rpm, 1800 psi (rotating), 2700 psi (stationary) at ambient temperatures on a 1-3/4-in. diameter shaft. This program is directed toward submarine pump applications.

The seal comprises the combination of a primary face-type mechanical seal, a secondary single floating ring seal, and a low-pressure face-type mechanical seal. An adjustable orifice regulates the pressure drop across the secondary seal, backed up by a relief valve.

Normal leakage is approximately 1 gpm which increases to 1-1/2 to 2 gpm on primary seal failure.

b - Byron-Jackson Company

A linear breakdown bushing development program receives primary emphasis at Byron-Jackson. Tests at 1800 rpm, 3000 psi, and 600 F, on a 6-in. diameter shaft are being conducted with leakage estimated at 75 gpm.

In addition a face-type mechanical seal development program with test facilities to 3600 rpm, 2500 psi, at ambient temperatures on a 2-in. diameter shaft is in progress. The seal comprises a series of arrangements of face-type mechanical seals with self-adjusting pressure distribution. Leakages are approximately 10 gph without external water injection. The design is not proven beyond 1500 psi.

c - Peerless Pumps

A combination seal development program with test facilities to 3600 rpm, 2500 psi, 500 F on shafts up to 4-1/2-in. diameter. The seal comprises four face-type mechanical seals arranged in two stages of pressure reduction with either internal or external bypasses. Leakages are approximately 1-2 gpm.

d - Bingham Pumps

A face-type mechanical seal development program with test facilities to 3600 rpm, 2500 psi, 600 F and a 4-5/8-in. diameter shaft. The seal
comprises a single face-type mechanical seal preceded by a throttle bushing to limit in-leakage to the pump casing. Leakages are approximately 1-2 gph. The design is presently limited to 1600 psi at 550 F.

e - Ingersoll-Rand Company

A floating-ring-type seal development program primarily for boiler feed pump service for sealing up to 4400 psi and 570 F. The seal comprises numerous floating-ring elements preceded by an initial throttle bushing and employing several stages of water injection for cooling and lubrication.

f - Allis-Chalmers Manufacturing Company

A floating-ring-type seal development program with undisclosed test conditions and objectives.

2 - Manufacturers Inactive in Seal Development

a - Westinghouse Electric Corporation

Westinghouse research and development is directed toward canned-motor pump technology.

b - General Electric Company

General Electric research and development is directed toward canned-motor pump technology.

c - Pacific Pumps

Pacific Pumps abandoned their early (1949) activities in seal research. At present they build controlled-leakage pumps with floating-ring seals for forced-circulation boilers.

d - C. H. Wheeler

C. H. Wheeler has unsuccessfully proposed pumps for nuclear service and has no current seal development program.

e - United Centrifugal Pumps

United Centrifugal Pumps has no current seal development program and is not engaged in nuclear activities.
3 - **Seal Manufacturers Engaged in Seal Development**

a - Borg-Warner

Borg-Warner manufactures seals for moderate pressure and temperature applications. Their development activities are directed toward high volume commercial applications.

b - Crane Packing Company

Crane Packing Company is engaged in a face-type mechanical seal development program with test facilities to 3600 rpm, 2500 psig, 300 F on a 4-3/8-in. diameter shaft. The seal employs an external heat exchanger to remove heat generated by the seal. No injection water is required. Results to date are encouraging since leakage rates are well below 1 gpm.

4 - **Current Manufacturers' Programs in Canned Pumps**

a - General Electric Company

General Electric has numerous development programs underway directed toward lowering the costs and improving the efficiency and service life of canned pumps both of the canned-rotor and wet-winding types.

b - Westinghouse Electric Corporation

Westinghouse engages in development work only in relation to current orders.
VI - CONCLUSIONS AND RECOMMENDATIONS

A - SHAFT SEAL DEVELOPMENTS

The large initial expense of main coolant pumps to satisfy zero leakage requirements was probably justifiable during the development stage of nuclear power over the past decade, when construction schedules were rigorous and hazard evaluations strict. However, with operational hazards better understood, consideration should be given to controlled leakage pump design in order to advance in the direction of commercially competitive nuclear power. The significant initial and operating cost incentive, as described earlier, fully justifies such a course of action.

Seals available today are not capable of operating at 2500 psig on the 6- to 7-in. shaft diameters required for a 40,000-gpm pump delivering 600°F water at 150 ft TDH with the minimum leakage, maximum reliability and service life demanded in utility operation. This then is the primary area for development. However, present-day pump manufacturers' seal development programs are not likely to achieve this objective for the following reasons:

a) Appropriated research funds are small.

b) Economic incentive is lacking due to small volume of anticipated business.

c) There is a tendency to postpone development until actual pump orders are placed.

d) There is a tendency to follow a single seal concept which appears momentarily advantageous.

e) No cross-fertilization of ideas occurs since each manufacturer hopes to achieve a proprietary advantage.

Accordingly an intensive research and development program should be initiated to accelerate the realization of a suitable pump shaft seal design by 1965 for plant operation in 1968.

In view of the many parallel seal developments currently being pursued and because it cannot be predicted with certainty that any one specific design will ultimately prove best, a coordinated course of development of the most promising seals is required. Accordingly we recommend that the program adopted conform to the following general requirements.
1 - **Over-All Program Objective**

The primary objective is to develop a face type mechanical seal within 4 to 5 years which is capable of operating continuously and reliably on a 6-in. diameter pump shaft at conditions of 2500 psig, 600 F, 1800 rpm, without external sources of injection water yet with controlled outboard leakage in the order of 1 gph. Alternate designs utilizing floating rings and/or breakdown bushings in combination with a face type seal will be considered if necessary to achieve the seal characteristics and life necessary for utility pumping application.

2 - **Required Facilities**

The facilities required during the first and second years may be quite modest. It is probable, however, that these facilities will be located in several shops. This will facilitate completion of the work on schedule and permit exploration of all attractive designs presently being investigated by individual manufacturers. The composite test facilities will include at least three variable speed seal test machines. The seal compartment design will permit 3000 psig, 750 F fluid conditions; shaft mountings of 6 in. diameter; varying water temperature and/or pressure at either constant or pulsating rates. In addition, means will be provided to create axial and radial shaft movements under load. The facilities will be complete with instrumentation to indicate and record all necessary variables. In addition to the test facilities, the following auxiliary facilities will be required at one or more locations:

a) Machine shop space and equipment to manufacture seal components for test.

b) A "clean" room for lapping, assembly, examinations and measurements of seal components.

c) A chemical and metallurgical laboratory.

3 - **Minimum Staff Required at a Single Facility**

- Research director
- Research scientist - engineer
- Seal test machine operator
- Skilled mechanic
- Chemist and/or metallurgist
- Designer/draftsman
4 - General Test Program

a) Design and construct mechanical seal components with varying geometric configurations to obtain test data concerning optimum seal face performance as related to its geometric configuration and also as related to shaft diameter, shaft speed, hydraulic pressure, spring closing pressure, clearance of rotating and stationary parts, etc.

b) Use alternate promising seal materials for the various seal designs per (a) above to determine, by test, the best materials for this service.

c) Arrange individual seal components into shaft seal systems for test in such a manner as to develop data concerning the effectiveness of multiple sealing faces in series, of interstage balancing and leak-off systems, of floating rings and breakdown bushings, etc.

d) Using the most attractive combinations of (a), (b) and (c) above, conduct initial proving tests in the seal test facility. Such tests shall consist of steady state runs, pulsing runs to simulate start-stop, pulsing runs to simulate rapid pressure and temperature transients on load and axial and radial expansion runs to determine over-all seal reliability. These initial proving tests will also include determinations as to seal suitability to withstand misalignments during erection, minor eccentricities as may be caused by wear and vibrations as may be inherent in system operation.

e) Upon completion of initial proving tests, those materials, designs and systems which prove most satisfactory will be individually subjected to a continuous 1000-hour test, at load, including tests similar to those conducted in the proving tests, but of a more rigorous nature and of longer duration.

It is expected that this work will be completed in two years. At that time it is expected that only a limited number of basic designs will justify further development and full-scale hot loop experiments. This final phase of the work can only be defined in detail after completion and evaluation of the described work.
5 - Program Costs - Approximate Estimate

a - Pre-Program Activities

Analysis and definition of the over-all program; detail layout of the test facilities; determination of the seal materials, designs and systems to be considered; outline of the test procedures and schedules; etc.; including preparation of detail inquiry specifications to permit obtaining research and development proposals from various manufacturers; and subsequent analysis of these proposals to select the seal researcher(s).

b - Research and Development - Years 1 and 2

Purchase and installation of seal test equipment; manufacture of seals and systems for test; manufacturers' research and development staff; miscellaneous materials, utilities, clerical, travel and living, and other "out-of-pocket" expenses; manufacturers' overheads, staff, shop, and laboratory facilities including researcher(s)' fees and program management expenses. Total estimated cost, first two years (a) and (b), $591,000. Minimum contingency, $109,000. Minimum suggested budget, $700,000.

6 - Potential Researcher(s)

The first two years of the program, when seal test machines only are utilized, may be accomplished in a seal or pump manufacturer's laboratory facilities. The latter half of the program requires the facilities of a pump manufacturer.

The following companies have sufficient laboratories and shop facilities to perform the work outlined for the first two years.

Bingham Pumps
Byron-Jackson Pumps Inc.
Crane Packing Company
Ingersoll-Rand Company
Peerless Pumps
Worthington Corporation

The Crane Packing Company is presently developing a face type mechanical seal requiring no injection water supply which has given encouraging results to date. Participation of several of the companies listed would lend diversity of approach and breadth of experience to enhance early successful completion of the work. Propositions for this
work should be solicited only after preparation of program specifications which clearly define the scope of work.

It is further recommended that the research and development program be placed under the cognizance of an Architect-Engineer Consultant who has had experience in the problems involved in controlled leakage pumps and their relations to nuclear reactor service. It is suggested that the Architect-Engineer Consultant be associated or engaged in metallurgical analysis and be fully competent to formulate the details of the program, review the findings and make recommendations as data are accumulated by various manufacturers under the sponsored program.

B - OTHER DEVELOPMENTAL AREAS

In addition to the recommended shaft seal development program, the following areas of possible cost reduction warrant full investigation:

1) Determine if more economical materials can be utilized in either canned motor and/or controlled leakage pumps maintaining high engineering standards, acceptable levels of water purity and plant safety. (This includes investigation of stainless clad components.)

2) Determine if inspection and testing procedures of major pump components can be altered or relaxed maintaining high engineering standards and component reliability.

3) Determine if canned motor pump manufacture and/or design can be modified to reduce high initial cost and/or increase motor efficiency.

In general, PWR systems utilize stainless steel materials throughout the primary loop. This selection is made, not for reasons of strength but principally to minimize the degree of corrosion products, irradiated or not, which can adversely affect system radiation levels and reactor core and steam generator heat transfer characteristics. Heretofore, estimated corrosion rates have been relatively conservative and no attempt has been made to accumulate data to consider substitution of low chrome alloys or stainless clad materials in the primary systems. Significant cost potential exists if alternate materials can be used. These potentials apply to both the canned motor and controlled leakage pump casing design. Since consideration of alternate materials is not a part of this study, it is suggested that the AEC undertake a separate feasibility study to determine the desirability of a research and development program investigating alternate
materials. At the same time it was emphasized during our study that a large portion of the cost of pumps is a result of the requirement to furnish nuclear grade castings rather than commercial castings. Interpretation of test readings has, heretofore, leaned heavily toward the conservative side. The cost incentive is significant since manufacturers stated that the nuclear grade casting costs are 30-50% greater than commercial grade castings. It is suggested that the feasibility study of alternate materials be expanded to include also a study of the desirability of a separate research and development program aimed toward relaxation of test standards. Such a program would principally involve comparison of pump casings with known defects such as internal shrinkage, surface cracks, small hot tears, etc., with casings that have been completely cleaned up to pass nuclear grade inspection. Such work would be of value not only in selection of pump materials and test standards but also for other costly elements in the primary circuit including piping, valves, reactor vessel, steam generator, pressurizer, etc.

Another area for development deals principally with the canned pump design. Some research is being carried out, particularly by the General Electric Company; however, their program is also limited by small research appropriations and limited business anticipation. This work should be expanded in those areas where significant cost savings can be realized. Again, a study should be conducted to determine the desirability of a separate research program and the specific areas for development. Such a program in parallel with the seal development may be justified in terms of military application or in the event the mechanical seal program yields unsatisfactory results. It appears that the following subjects could be part of this separate research and development program.

1) Investigation of canning materials to reduce electrical losses thereby increasing motor efficiency. Present canned motors are being designed with efficiencies of 64 - 69% compared to standard frame size motors where efficiencies approximate 92 - 95%. The canned motor efficiencies can be segregated into the following losses:
   a) Air Gap Loss (Canning Material) - 50%
   b) Hydraulic Loss (Fluid Rotation) - 35%
   c) Normal Motor Loss (Remainder) - 15%
It can readily be seen that the Air Gap (canning materials) accounts for the greater loss and this is definitely an area that presents a chance of betterment. As an example of this it is believed that a change from Inconel material to Hasteloy could increase efficiency possibly 2 - 3% which is a substantial gain. The use of nonmetallic canning material is also a possibility.

2) **Further development of water lubricated bearings and bearing materials with emphasis on corrosion resistance and high-temperature applications.**

At the present time most canned pumps employ Graphitar (graphite and carbon with suitable binder) material for both thrust and journal water lubricated bearing service. This material cannot presently stand up under temperatures over about 300 F due to breakdown of binding material. Therefore, bearings are limited in location within the pump and have a direct relation to pump design. There is also much to be done on water lubricated bearings from an operational standpoint to promote reliability.

3) **Development of wet motors with view toward elimination of canning material.**

Wet motors today have been quite successful on boiler circulating service. Where pumped fluid comes in contact with motor windings, no deleterious effects have been observed. Investigation of irradiation on motor insulation should be made.

4) **Study and test gasketed type closures to eliminate seal welds but maintain comparable degree of safety.**

5) **Study of single-pressure vessel concept to eliminate double heavy wall shell around spiral-wound heat exchanger.**
APPENDIX

Inquiry Specification Sample
Comparison of Pump Costs
INVESTIGATION OF TYPE OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

PUMP MANUFACTURER DATA SHEET

I - GENERAL

A - Company Name & Year Formed _________________________________

1 - Locations
   a - Principal Office _______________________________________
   b - Engineering Office(s) ________________________________

   c - Sales Office(s) _______________________________________
       (Enter principal & local pump product representation)

   d - Shop(s) and Items Produced ______________________________

   e - Parts & Service Facilities (location) _______________________

2 - Approx Pump Sales Volume in Dollars _________________________

3 - Pump Types & Sizes Produced _______________________________

4 - Source of Pump Components
    (Manufactured by Subject Co or Procured from Other Sources - List Principal Sources for Basic Items Including Seals)
I - GENERAL (cont'd)

5 - Pump Shop(s) (Manufacturer to submit one Page 2 for each Pump Shop)
   a - Shop Name and Address ____________________________________________

6 - Test Facilities at This Pump Shop
   a - Component Testing
       (Material Testing)  (Cite applicable codes and list items tested with test(s) actually performed such as radiographic and fluid penetration inspection of castings and forgings, weld tests, ultrasonic testing of pressure parts, hydrostatic or helium leak tests of casings and electrical tests of motor windings including dialectric tests, insulation resistance and electrical balance tests).
   b - Performance Testing
       1) Max Flow, gpm, cold
       2) Hot Load Testing: gpm to °F
       3) What results are obtained (TIH, Elec Power Input, Overall Eff, Motor Current, Power Factor, etc.)
       4) Vibration Testing (Flow Throttled to Specified Limit of Vibration. What Results are Obtained?)
       5) Seal Testing?
# INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

## PUMP MANUFACTURER DATA SHEET

### CANNED MOTOR DESIGN

## II - PUMPS SOLD

*Manufacturer to submit detailed sheets for each pump size and type sold*

### A - Customer

*Name, Address & Year of Installation*

### 1 - Application (Use)

#### a - Design Conditions

1. Max. Ambient Temp ($^\circ$F)
2. Max. Disch Press. (psig)
3. Fluid Pumped
4. Cooling Medium
5. Incident Radiation

### 2 - Pump Specifications

#### a - Type

#### b - Capacity (gpm)

#### c - HP and Eff at 100%, 75%, 50%, Other

#### d - Operating Pressure (psig)

#### e - Suction Pressure (psig)

#### f - TDH - ft

#### g - Minimum Req'd NPSH (ft)

#### h - Available NPSH (ft)

#### i - Vapor Press. on Suction Max. psig

#### j - Vapor Press. on Suction Min. psig

#### k - Max. & Min. Flows GPM

#### l - Fluid Temperature ($^\circ$F)

#### m - Rotation & Speed (rpm)

#### n - Motor

*Type & Manufacturer*  
*Duty*  
*WR$^2$ (complete rotating element)*
## II - PUMPS SOLD

### 2 - Pump Specifications (Cont'd)

<table>
<thead>
<tr>
<th>Letter</th>
<th>Description</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>o</td>
<td>Estimated Quantity pumped on motor trip due to WR^2</td>
<td></td>
</tr>
<tr>
<td>p</td>
<td>Fluid Operating Temp. Variation Allowed</td>
<td>( \text{F} ) to ( \text{F} )</td>
</tr>
<tr>
<td>q</td>
<td>Operating Pressure Variation Allowed</td>
<td>( \text{lb (psig)} ) to ( \text{lb (psig)} )</td>
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<tr>
<td>r</td>
<td>Operation During Transient Conditions</td>
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<tr>
<td>s</td>
<td>Flow Surface Finish ( \text{rms average) Casing}</td>
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<tr>
<td>t</td>
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<td>2. Impellers</td>
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<td>3. Shafting</td>
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<td>4. Bearings</td>
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</tr>
<tr>
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<td>b</td>
<td>Type</td>
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<td>c</td>
<td>Diffuser or Volute</td>
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<td>d</td>
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<td>g</td>
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<td>h</td>
<td>Motor Terminal Connection (Type)</td>
<td></td>
</tr>
<tr>
<td>i</td>
<td>Grounding Pads</td>
<td></td>
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</tbody>
</table>
II - PUMPS SOLD

3 - Construction Details (Cont'd)
   i - Monitoring Points
       (What kind, Where?)

   j - Type of Gaskets Used

   k - Mounting

   l - Is Reverse Rotation Check Provided. (Ratchet or other Device)

   m - Welding Procedures

   n - What Provision is Made to Insure Correct Rotation

4 - Approximate Initial Cost

5 - Approximate Installation Cost

6 - Guarantees

7 - Characteristic Curve (Attach.)

8 - Cooling Water Requirements (gpm) (at 50%, 75% and 100% load)
     a - Size & Cost of CW Pump
         If Supplied as External Unit

     b - Cooling Water Press. Req., psig

     c - Cooling Water Temp. Req., F Max.

9 - Shop Tests

10 - Field Tests

11 - Experience Record
### PUMPS SOLD

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<thead>
<tr>
<th>Number</th>
<th>Description</th>
<th>Data Sheet</th>
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<tr>
<td>11</td>
<td>Cross Section Drawing (Attach)</td>
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<td>12</td>
<td>Outline Drawing (Attach)</td>
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<td>13</td>
<td>Manufacturer's Recommended Servicing - Job Site</td>
<td></td>
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<tr>
<td></td>
<td>a - Pump</td>
<td></td>
</tr>
<tr>
<td></td>
<td>b - Motor</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Manufacturer's Recommended Servicing - Factory</td>
<td></td>
</tr>
<tr>
<td></td>
<td>a - Pump</td>
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<td>15</td>
<td>Expected Annual Maintenance Costs</td>
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</tr>
<tr>
<td>16</td>
<td>Recommended Spare Parts Req'd for First 5 Years of Operation &amp; Approx. Cost.</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>Servicing and Dismantling Tools or Fixtures Req'd &amp; Approximate Cost.</td>
<td></td>
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</tbody>
</table>
### INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

**PUMP MANUFACTURER DATA SHEET**

**CONTROLLED LEAKAGE DESIGN**

### III - PUMPS SOLD

(Manufacturer to submit detailed sheets for each pump size & type sold)

**A - Customer**

(Name, Address & Year of Installation)

**1 - Application (Use)**

**a - Design Conditions**

1. Max Ambient Temp ($^\circ$F)

2. Max Disch Press (psig)

3. Fluid Pumped

4. Cooling Medium

5. Incident Radiation

**2 - Pump Specifications**

**a - Type**

**b - Capacity (gpm)**

**c - HP and Eff at 100%**

75%

50%

Other

**d - Operating Pressure (psig)**

**e - Suction Pressure (psig)**

**f - TDH (ft)**

**g - Minimum Req'd NPSH (ft)**

**h - Available NPSH (ft)**

**i - Vapor Press on Suction Max psig**

**j - Vapor Press on Suction Min psig**

**k - Min & Max Flows (gpm)**

**l - Fluid Temperature ($^\circ$F)**

**m - Rotation & Speed (rpm)**
### III - PUMPS SOLD

#### 2 - Pump Specifications (Cont'd)

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<table>
<thead>
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<tbody>
<tr>
<td>n</td>
<td>Motor</td>
</tr>
<tr>
<td></td>
<td>Type &amp; Manufacturer</td>
</tr>
<tr>
<td></td>
<td>Duty</td>
</tr>
<tr>
<td></td>
<td>WR² (complete rotating element)</td>
</tr>
<tr>
<td>o</td>
<td>Estimated Quantity Pumped in Motor Trip due to WR²</td>
</tr>
<tr>
<td>p</td>
<td>Fluid Operating Temperature Variation Allowed</td>
</tr>
<tr>
<td></td>
<td>F to F</td>
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<tr>
<td>q</td>
<td>Operating Pressure Variation Allowed</td>
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<td></td>
<td>lb (psig) to lb (psig)</td>
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<td>r</td>
<td>Operation During Transient Conditions</td>
</tr>
<tr>
<td></td>
<td>1. Temp Rate of Change Limit</td>
</tr>
<tr>
<td></td>
<td>Casing</td>
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<td></td>
<td>Impeller</td>
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<td>t</td>
<td>Materials</td>
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<td>1. Casings</td>
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#### 3 - Construction Details

<p>| | |</p>
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<td>d</td>
<td>Vertical or Horizontal</td>
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### III - PUMPS SOLD

#### 3 - Construction Details (Cont'd)

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<tr>
<td><strong>e</strong> - Casing - Welded or Flanged and Bolted (Applicable Code)</td>
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<tr>
<td><strong>f</strong> - Piping Connections</td>
<td></td>
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<tr>
<td>Suction (Size, Sched)</td>
<td></td>
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<tr>
<td>Discharge (Size, Sched)</td>
<td></td>
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<tr>
<td>Other</td>
<td></td>
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<tr>
<td><strong>g</strong> - Motor Terminal Connection (Type)</td>
<td></td>
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<td><strong>h</strong> - Grounding Pads</td>
<td></td>
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<tr>
<td><strong>i</strong> - Monitoring Points (What Kind, Where?)</td>
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<td><strong>k</strong> - Mounting</td>
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<td><strong>l</strong> - Is Rotation Check Provided (Ratchet or other Device)</td>
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#### 4 - Approximate Initial Cost

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#### 5 - Approximate Installation Cost

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#### 6 - Guarantees

<p>| |</p>
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#### 7 - Characteristic Curve (Attach)

<p>| |</p>
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</table>

#### 8 - Cooling Water Requirements (gpm) (at 50%, 75% and 100% load)

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<tr>
<th></th>
<th></th>
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</thead>
<tbody>
<tr>
<td><strong>a</strong> - Size &amp; Cost of CW Pump If Supplied as External Unit</td>
<td></td>
<td></td>
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</tbody>
</table>

<p>| |</p>
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</table>
INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

PUMP MANUFACTURER DATA SHEET
CONTROLLED LEAKAGE DESIGN

III - PUMPS SOLD

8 - Cooling Water Requirements (gpm) (at 50%, 75% and 100% load) (Cont'd)
   b - Cooling Water Press (psig)______________________________
   c - Cooling Water Temp Req (F max)__________________________

9 - Field Tests_____________________________________________

10 - Experience Record________________________________________

11 - Gross Section Drawing (Attach)____________________________

12 - Outline Drawing (Attach)_______________________________

13 - Manufacturer's Recommended Servicing - Jobsite
   a - Pump_______________________________________________
   b - Motor_____________________________________________

14 - Manufacturer's Recommended Servicing - Factory
   a - Pump_______________________________________________
   b - Motor_____________________________________________

15 - Expected Annual Maintenance Costs_________________________

16 - Recommended Spare Parts Required for First 5 Years of Operation & Approx Cost
# INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

**PUMP MANUFACTURER DATA SHEET**

**CONTROLLED LEAKAGE DESIGN**

## III - PUMP SOLD

17 - Servicing & Dismantling
   Tools or Fixtures Required
   & Approx Cost

18 - Type of Seal Arrangement
    (Explain Design such as
    Mech Seal, Labyrinth,
    Injection or Combination)

19 - Source of Seal Components
    (Own mfg or Name Supplier)

20 - State Pressure Temperature
     Limitations on Seal Components

21 - Outline or Attach Curves of
     Leakage Guarantee (New Seals)

22 - Outline or Attach Curves of
     Leakage Guarantee on Basis
     of Worn Seals.

23 - State Required Quantity in
     gpm, Pressure & Max Temperature
     of Seal Water (If Required)

24 - Give Estimate and/or Guarantee
     of Max Leakage if Seal Fails

25 - Give Estimate and/or Guarantee
     on Minimum Seal Life

26 - Attach Detailed Drawings of
     Seal Assembly and Flow Diagram
     of Seal Piping
INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

MANUFACTURING CAPABILITY

IV - PUMPING PRODUCTS AVAILABLE
(Obtain List and Attach)

A - Limiting Conditions Affecting Current Design

1 - Hydraulic Limitations (gpm, NPSH, TMH, etc.)
   a - Canned Motor Design ____________________________
       ____________________________________________

       ____________________________
       ____________________________________________

   b - Controlled Leakage Design ______________________
       ____________________________________________

       ____________________________
       ____________________________________________

2 - Sealing Limitations (Press, Temp, Diameters, Materials, etc.)
   a - Canned Motor Design ____________________________
       ____________________________________________

       ____________________________
       ____________________________________________

   b - Controlled Leakage Design ______________________
       ____________________________________________

       ____________________________
       ____________________________________________

3 - Other Limitations (R&D costs, order repeatability, etc?)

       ____________________________________________
       ____________________________________________

       ____________________________
       ____________________________________________

       ____________________________________________
       ____________________________________________
INVESTIGATION OF TYPES OF SEALS FOR MAIN COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

IV - (cont'd)
B - Pumps Available, Projected to 1965 Based Upon Manufacturers Present Programs

1 - Hydraulic Capability
   a - Canned Motor Design

   b - Controlled Leakage Design

2 - Sealing Capability & Materials Development
   a - Canned Motor Design

   b - Controlled Leakage Design

C - Pumps Available Projected to 1965 Based Upon a Sponsored Research and Development Program Beyond Manufacturers Present Programs

1 - Hydraulic Capability
   a - Canned Motor Design

   b - Controlled Leakage Design

2 - Sealing Capability and Materials Development
   a - Canned Motor Design
b - Controlled Leakage Design

D - New Pump Manufacturing or Pump Test Facilities Under Construction or Projected
1 -
2 -
3 -
4 -
5 -
INVESTIGATION OF TYPES OF SEALS FOR MAIN
COOLANT PUMPS FOR LARGE NUCLEAR PLANTS

V - RESEARCH AND DEVELOPMENT PROGRAMS

A - Current Program(s) (manufacturer to outline present plans)

B - Future Program(s) Planned to 1967 (Manufacturer to outline projected plans)
C - Suggested Programs Based Upon a Sponsored Development Program
(manufacturer to outline program he would undertake under sponsorship)
VI - ECONOMIC COMPARISONS

Manufacturers' estimated cost for a pump with the following general characteristics now and projected into 1965, taking into effect results of Manufacturers' R&D programs and general advances and changes in design expected.

*Type - Manufacturer to specify

<table>
<thead>
<tr>
<th>Operating Pressure (psig)</th>
<th>2500 lb</th>
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</thead>
<tbody>
<tr>
<td>Fluid Pumped</td>
<td>Demineralized Water</td>
</tr>
<tr>
<td>Temperature of Fluid</td>
<td>625 F</td>
</tr>
<tr>
<td>Pump Materials</td>
<td>Stainless Steel</td>
</tr>
<tr>
<td>Piping End Connections</td>
<td>Butt Welding</td>
</tr>
<tr>
<td>Pump Capacity</td>
<td>40,000 gpm</td>
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<tr>
<td>TDH</td>
<td>150 ft</td>
</tr>
<tr>
<td>NPSH</td>
<td>30 ft</td>
</tr>
</tbody>
</table>

Motor - Open, Drip Proof, Single Speed

Type
Induction

Duty
Continuous

Transient Requirements

Pump shall be capable of running continuously at full speed without damage with restricted flow. The fluid temperature during this operation will vary between 50 F and 600 F. Pump shall also be capable of starting against a shutoff head with fluid temperatures between 50 F and 625 F and shall be capable of operating successfully in parallel with one or more duplicate pumps over their full range of capacity.

During the following transient conditions, the pump shall be capable of operating without damage:

a - Fluid pressure will vary between 2250 psig and 2750 psig during normal operation.

b - Fluid temperatures will vary between 50 F and 625 F during startup and shutdown at a rate not exceeding 100 F/hr at shutdown and 60 F/hr at startup for a total of 100 transients for each operation.

<table>
<thead>
<tr>
<th>Cost (including motor)</th>
<th>Current</th>
<th>1965</th>
</tr>
</thead>
<tbody>
<tr>
<td>Canned Rotor</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Controlled Leakage</td>
<td></td>
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</tr>
</tbody>
</table>

* Manufacturer is requested to submit a separate quotation for each basic design and to quote separately any differential costs for each variation in his basic design.
VIII - **RECOMMENDED SEALS (BY MANUFACTURER)**

1 - Type and Size

2 - Manufacturer

3 - Cost

4 - Expected Performance
   (Maximum rate of leakage)

5 - Servicing required

6 - Expected life

7 - Maintenance problems expected

8 - Construction (Attach Drawing Bill of Materials)

9 - Limitations
   (Temperature, shift size, etc)

10 - General (Any data on these seals available)
The pump Inquiry Specification (pages 17 and 18) included a preliminary pump specification for the purpose of obtaining estimated pump prices typical of 1960 cost levels, and estimated pump prices projected to 1965 based on 1960 dollars.

The manufacturers interviewed were unable to submit firm quotations, or any quotations at all in some cases, because time did not permit a detailed engineering study and cost estimate. The prices tabulated herein are therefore the best available estimates at this time, based upon their manufacturing experience and familiarity with the general requirements of nuclear service.

Manufacturers' names are omitted from this tabulation due to the tentative nature of their responses.
## ESTIMATED PUMP COSTS, EACH (INCLUDING MOTOR)

### CONTROLLED LEAKAGE PUMP

<table>
<thead>
<tr>
<th>Mfr</th>
<th>Pump and Motor</th>
<th>Installation</th>
<th>Supervision of Erection</th>
<th>Starting and Testing</th>
<th>Total Cost</th>
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<tbody>
<tr>
<td>A</td>
<td>$160,000</td>
<td>-</td>
<td>$9,000</td>
<td>$700</td>
<td>$300</td>
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<tr>
<td>B</td>
<td>200,000</td>
<td>-</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
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<tr>
<td>C</td>
<td>225,000</td>
<td>-</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
</tr>
<tr>
<td>D</td>
<td>228,000</td>
<td>$275,000</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
</tr>
<tr>
<td>E</td>
<td>-</td>
<td>250,000</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
</tr>
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<td>F</td>
<td>255,000</td>
<td>318,000</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
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<tr>
<td>G</td>
<td>202,000</td>
<td>248,000</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
</tr>
<tr>
<td>H</td>
<td>No Bid</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
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### CANNED PUMPS

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<tr>
<th>Mfr</th>
<th>1960</th>
<th>1965</th>
<th>Installation</th>
<th>Supervision of Erection</th>
<th>Starting and Testing</th>
<th>Total Cost</th>
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<tr>
<td>X</td>
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<td>$350,000</td>
<td>$9,000</td>
<td>$700</td>
<td>$300</td>
<td>$535,000&lt;sup&gt;c/&lt;/sup&gt;</td>
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<tr>
<td>Y</td>
<td>600,000</td>
<td>-</td>
<td>9,000</td>
<td>700</td>
<td>300</td>
<td>610,000</td>
</tr>
</tbody>
</table>

**a/** Manufacturers D, F & G assumed an escalation factor plus stricter interpretation of nuclear service pump standards.

**b/** Manufacturer E was unable to estimate a pump today. His future price is based upon a successful seal development program and hydraulic model tests. As an alternate he quoted on (3) 14,000 gpm pumps.

**c/** Manufacturer X based his future price on a successful series of improvements in the technology of canned pumps.
BIBLIOGRAPHY

1 - ASME Paper No. 55-SA-42, Water Lubricated Bearing Study
2 - Power Engineering, September 1957, Boiler Feed Pump Stuffing Boxes
3 - Power Engineering, May 1956, Boiler Feed Pump Stuffing Boxes
4 - Westinghouse Technical Bulletin 57-565, Zero Leakage Canned Motor Pump
5 - Westinghouse Technical Bulletin 57-550, Zero Leakage Canned Motor Pump
6 - Centrifugal Circulating Pump and Auxiliary Pumps for Atomic Power Plants, Carl Blom, Byron Jackson Company
7 - Dynamic Sealing, Theory and Practice, Koppers Company
8 - Packing and Mechanical Seals, Crane Packing Company
10 - Engineering Problems of Power Reactors, ASME Paper No. 55-SA-77
11 - Gland Sealing in High Pressure Boiler Feed Pumps, H. L. Ross, Allis-Chalmers Manufacturing Company
12 - Description of the Shipping Port Atomic Power Station, Westinghouse Electric Corporation for AEC, WAPD-PWR-970
13 - Instruction Book Main Coolant Pump, PWR, Shipping Port Power Station