Chapter Eight

PROCESS EQUIPMENT

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8.1 Introduction

Two of the largest and most complex items of process equipment, the steam generators and the heat transport pumps, are considered in this chapter. While nuclear steam generators generally consist of heat exchangers in conjunction with steam separation equipment, the internal thermohydraulics is quite complex, as is the detail design. Tube bundle design, steam separation equipment design, internal thermohydraulics and a historical summary are covered in separate sections. For heat transport pumps, the hydraulics and single phase performance aspects is given, as are construction and design features of the pump/motor assembly. Shaft seals used in these pumps and abnormal operation considerations (such as impaired services to the pumps and post-LOCA operation) are briefly covered.

8.2 Steam Generators

8.2.1 Historical Summary & Types

In the first CANDU Nuclear Power Plant NPD (Nuclear Power Demonstration) a single steam generator produces steam to make 20 MW(e) of power. This steam generator is horizontal, with a steam drum above the bundle as shown on Figure 1. This is the only CANDU steam generator with a horizontal tube bundle design; the tube material is Inconel-600(*).

This steam generator has performed well, with twenty years of operation behind it. There has been some tube wall deterioration and recently, the secondary side was cleaned. Copper bearing deposits were removed and the thermal performance of the unit has been restored.

By way of comparison, the Shippingport plant in the United States (commissioned before NPD) has steam generators that are very similar to NPD. They have a horizontal tube bundle as is shown in Figure 2. The Shippingport plant has experienced a number of tube failures which have been of the nature of tube leakage at internal support plates and these failures have, in many cases, been traced to corrosion.

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* Inconel 600 is used herein to denote nickel-chromium-iron alloy specified in ASME SB-163, with restrictive composition. Similarly, where Monel-400 and Incoloy 800 are used herein, they denote nickel-copper alloy and nickel-iron-chromium alloy, respectively, as specified in ASME SB-163, with restrictive composition.

Corrosion problems were known or suspected by AECL when we embarked on the design of a full-scale plant in the 1950's. Our extensive tube corrosion research effort started then and continues today.

Early design effort produced the 200 MW(e) Douglas Point and Rajhastan designs which have been quite successful. The arrangement of the Douglas Point steam generators is shown on Figures 3 and 4. These steam generators were the first to incorporate an internal preheater. Monel-400(*) alloy was used instead of Inconel-600 as the tube material, because of its economic advantages, its better thermal conductivity, and its better resistance to some forms of corrosion attack.

At the time AECL designed the Douglas Point steam generators, there was concern about the thermal fatigue stresses that might result from a temperature difference across a single tube sheet. This concern results in a U-tube and U-shell design, oriented vertically, with separate tube sheets for the inlet and outlet ends of the tubes. A large number of steam generators were used, connected to common steam drums. One of the advantages of using a large number of small heat exchangers is that we were able to perform a full scale test on one of the vessels to thoroughly investigate its performance.

Each of the four 515 MW(e) reactors at Pickering have a steam generator design that is now well known. These were the first CANDU steam generators having a "light-bulb" design. The general arrangement of the Pickering steam generators is shown on Figures 5 and 6.

The major features of the Pickering design are:

1. The tube bundle is oriented vertically and there is only one tube sheet.
2. The tubes are Monel-400.
3. The steam drum is located on top of the secondary side shell.
4. The steam generator contains an integral preheater.

These steam generators are the most reliable and successful that have ever been built, world-wide. There has been only one tube leak in the ten year history, which was due to a random manufacturing defect in the tube. No significant loss of production has resulted from any cause related to the steam generators.

The Bruce 'A steam generators came next. (These are shown on Figure 7.) There are three main differences in the Bruce 'A' steam generators relative to the Pickering steam generators, namely:

1. A common steam drum is used for four steam generators in a manner similar to that of Douglas Point. A common drum was used in order to simplify the steam drum water level control design requirements.

2. A separate preheater vessel is used to accommodate the two temperature reactor core that is used in Bruce 'A'.

3. The tube material is Inconel-600.

The steam generators for Bruce 'B' will be of the "light-bulb" type, but will retain the separate preheater as for Bruce 'A'.

The CANDU 600 and CANDU 950 steam generators are of the vertical U-tube and natural circulation type, with preheaters and integral steam drums (see Figures 8 and 9). These are similar to the Pickering concept, except that the tubing is Incoloy-800 rather than Monel. This material is preferred to Inconel-600, to which it is very similar, because it is less expensive and it avoids the pure water cracking phenomena to which nickel-chromium-iron alloy (Inconel-600) has shown susceptibility.

Some of the pertinent design parameters of the CANDU 600 Steam Generators as compared to previous steam generators are given in Table 1. There are 4-CANDU steam generators per 600 MW reactor (i.e. about 150 MWe net per vessel). The steam generators are approximately 2.7 m in diameter (steam drum) and 19.3 m long. The shipping weight of each vessel is approximately 230 tonnes and, hence, does not pose any transportation problems; these steam generators can be moved by land, rail or sea transportation.

TABLE 1: CANDU STEAM GENERATOR PHYSICAL PARAMETERS

	Pickering A	Bruce A		CANDU 600
		Preheater	Steam Generator	
Overall height (m)	14.2	5.5	14.9	19.1
Heat transfer area per vessel (m^2)	1858	829	2415	3177
Tubing: material	M-400	I-600	I-600	I-800
no. U-tubes/vessel	2600	2900	4200	3550
o.d. (mm)	12.60	12.954	12.954	15.9
wall thickness (mm)	1.24	1.1303	1.1303	1.13
Tubesheet: diameter (m)	1.83	2.18	2.52	2.77
thickness (m)	0.27	0.32	0.37	0.38
Steam drum diameter (m)	2.5	----	2.86	4.01
Steam drum wall thickness (mm)	58.7	----	58.7	72.9
Steam generator diameter (m)	1.73	2.14	2.39	2.66
Steam generator wall thickness (mm)	41.2	67.6	50.0	49.3

8.2.2 Tube Bundle Design

General

All recent CANDU steam generators have preheaters and evaporator sections; no superheaters have been used. For the CANDU 600 and CANDU 950 steam generators, three main thermal zones are considered as shown on the heatload diagram (Figure 10). The condensing zone is the region of the D_2O inlet leg where D_2O with quality is condensed to saturated D_2O. The central zone is the main evaporator zone. The preheating zone is, in effect, a liquid-to-liquid heat exchanger in which the incoming feedwater is heated (in a counterflow pattern) by the cold leg D_2O flow. The use of a preheater improves the thermal effectivenss of the steam generating unit and hence reduces D_2O inventory.

The preheating zone effectiveness and the steam pressure attainable are limited by the "pinch point" experienced at the outlet from the preheater. In selecting the pinch point temperature difference, a compromise is made between steam generator surface (and hence D_2O inventory) and steam pressure.

Typical CANDU 600 steam generators conditions are given below in Table 2.

TABLE 2

TYPICAL CANDU 600 STEAM GENERATOR CONDITIONS

Steam Output (total of 4 steam generators)	3.72×10^6 Kg/hr (8.20×10^6 lb/hr)
Steam Pressure	4.70 MPa(a) (681 psia)
Steam Temperature	260°C (500°F)
Feedwater Inlet Temperature	187°C (368°F)
Heavy Water Inlet Quality (by weight)	4.4%
Heavy Water Outlet Temperature	266°C (511°F)
Moisture in Steam	0.25% (maximum)
Heat Transfer Rate (heat to 4 steam generators)	2064 MW(th)

Integral Preheater Design Considerations

To provide an integral preheater on the cold leg of the U-tubes, CANDU steam generator preheaters utilize crossflow with baffles of either single segmental or double segmental type to maximize the thermal effectiveness of the preheater while, at the same time, minimizing flow induced vibration. Consideration must be given to flow conditions near the top of the preheater where feedwater flow approaches saturation.

Careful consideration must also be given to the area where the incoming feedwater enters through the thick shell to the tube bundle. In CANDU steam generators with integral preheaters, the feedwater flow enters through a thermal sleeve in the feedwater nozzle to a feedwater distribution box. This box is usually provided with an orifice to direct flow uniformly to inlet ports, where inlet flow velocities to the tubes are considerably lowered from velocities in the feedwater piping.

Since the integral preheater also involves the introduction of relatively cold feedwater into the steam generator in the region just above the tubesheet, careful consideration must be given to the design in this region. CANDU steam generators use either a thermal plate or a thermal shield layer on the tubesheet and this area is subject to detailed analysis to ensure acceptable tube and tubesheet stress levels.

Evaporator Section Design Considerations

The design of nuclear steam generator tube bundles involves quite a number of considerations: the thermal surface must be selected as well as tube size, the number of tubes per unit, the tube pitch, and the tube length.

For CANDU steam generators, relatively small tubes are used, largely for D_2O inventory reasons. Both 12.7 mm (1/2") and 15.9 mm (5/8") outside diameter tubes have been used, with the latter size favoured for our recent units. Selection of the bundle shape, once the surface area required has been determined, involves a number of compromises. Long, narrow tube bundles are the most economic to build but there usually are building height restrictions on the length of the steam generator as well as limits on primary side pressure drop. Selection of tube pitch involves a compromise between bundle diameter (and hence economics) and thermal hydraulic considerations in the tube bundle.

In the selection of the tube bundle geometry, the designer must be aware of the experience that has shown that having good thermal/hydraulic conditions on the secondary side of the tube bundle in vertical, natural circulation nuclear steam generators is one of the key elements in reducing tube failures and maintaining operational availability. The region in the tube bundle just above the tubesheet historically has been a problem area where tube failures have occurred. Poor thermal hydraulic conditions in this region have been one of the contributing factors for these tube failures. The problems have often occurred on steam generators with large tube nests where little recirculating water reaches the tubes furthest from the downcomer inlet; hence, regions of high steam quality have occurred just above the tubesheet. This is also the region in the steam generator evaporator zone where heat fluxes are the highest.

High steam quality and low flow velocity have resulted in sludge accumulation and associated stress corrosion cracking of the tubes due to local chemistry excursions in the sludge. Proper distribution and penetration of the downcomer water into the tube bundle provides optimum thermal hydraulic conditions.

The number, size and geometry of CANDU steam generators has been selected such that tube bundle diameters have remained relatively small while reactor unit size has grown, as given below in Table 3.

Table 3 - CANDU Steam Generator Sizes

	Pickering A Pickering B	Bruce A Bruce B	CANDU 600	Darlington A	CANDU 950
Total Steam Generator Heat Transfer (MW(th))	1660	2665*	2064	2657	3392
No. of Steam Generators per Reactor	12	8	4	4	8
Tube Bundle Diameter (m)	1.47	2.01	2.22	2.66	2.08

*Steam generators only. An additional 444 MW(th) are transferred in separate preheaters.

It should be noted that all CANDU steam generators are required to have relatively high recirculation ratios. AECL practice has been to specify a minimum recirculation ratio of 5. A further significant feature of CANDU steam generators is that the hot leg heat fluxes are relatively low compared to many PWR steam generator designs.

Because of the relatively small tube bundle diameter of CANDU steam generators compared to larger units, the incoming recirculating water to the evaporator section crosses relatively few tube rows. This, in combination with the high recirculating water flow rates entering the bundle and the low heat fluxes, allows low steam quality levels to be achieved in the evaporator region just above the tubesheet.

Designs are carefully reviewed regarding the penetration and distribution of downcomer flow into the tube nest, and special measures are taken, including flow testing, to provide assurance of acceptable flow patterns.

Evaporator Section Tube Supports

Another area that is given very detailed attention is the natural circulation loop and its thermal hydraulic considerations. As the height of the steam generators is limited by space considerations, the differential head available to provide high secondary side circulation ratios is also limited. Hence, the design should minimize the secondary side pressure drop through the evaporator zone. One of the largest of these losses is due to expansion and contraction of the two-phase flow through the numerous tube supports. Hence, tube supports in both the parallel flow zones and in the U-bend region are carefully selected to obtain the highest possible porosity allowed by mechanical strength requirements or manufacturing considerations. The high porosity significantly reduces the pressure drop across the tube supports and allows higher recirculation rates to be obtained. In addition to providing high porosity, tube supports must have large open-flow regions around each tube to minimize the risk of possible deposit accumulation in the tube holes around the tubes, and minimize the risk of tube denting.

All CANDU steam generators starting with Pickering 'A' have met these requirements. The Pickering 'A' steam generators were supplied with a lattice bar type of tube support in which alternate slotted rectangular bars are used to form a tube support (see Figure 11).

Pickering 'B', Bruce 'A' and 'B' and a number of CANDU 600 plants have steam generators supplied with tube supports that have a trilobar broached tube hole in the support plate for every tube (see Figure 12). This design has three open flow passages around each tube, supports each tube at three contact areas, and gives high porosity to the longitudinal two-phase boiling fluid flowing vertically upward over the outside of the tubing.

Another type of tube support that has been used is a grid-type support. In this arrangement, the supports are fabricated from a series of formed strips, which are welded together to obtain a rigid disk of close tolerances.

Since the U-bend region is subject to high two-phase cross flow velocities and is the highest quality region in the tube bundle, AECL requires that the U-bend tube supports be specially designed, fabricated and assembled to minimize the possibility of concentration of any chemicals present in the water in this region of the tube bundle and yet to provide an adequate number of support locations on the U-bend portion of each tube. Hence, the number of supports provided is partially dependent upon the bend radius involved, with the largest bend radius having, as a minimum, support at the 45° and 90° locations of the U-bends.

8.2.3 Steam Separation Equipment

The steam separation equipment in nuclear steam generators is physically located in the steam drum region (see Figures 8 & 9). The function of this equipment is to separate the water and steam phases to allow essentially dry steam to exit from the steam generator steam nozzle and to have steam-free water return to the secondary side natural circulation loop.

The steam separation equipment influences the operation and performance of a nuclear steam generator via pressure drop, carryover and carryunder characteristics.

Carryover is the moisture entrained in the exiting steam flow and if it is excessive (beyond the design target limit) it can result in decreased plant efficiency and may lead to premature turbine blade failure.

Carryunder is the steam entrained in the recirculating water flow. If it exceeds design targets, it has a potential for decreasing the recirculating water flow rate (and hence, the recirculation ratio) and it may subsequently adversely affect the operating performance of the steam generator. If excessive carryunder exists, the differential pressure that provides the natural circulation driving force (see Section 8.2.4) is reduced.

The pressure drop of the steam separation equipment is minimized to the extent possible by design. The influence of this pressure drop on natural circulation loop hydraulics and on heat transfer surface (via the "submergence" effect) is explained in Section 8.2.4.

The selection and design of the steam separation equipment for nuclear steam generators recognizes the above three performance characteristics. Two-phase thermal/hydraulic technology is not yet at a stage where the design of this equipment can be done "on paper" using analytical predictions. While there is understanding of the parameters which affect separation equipment performance, design relies on experience in actual units and on test programs in air/water and steam/water. Due to test facility limitations and economics, various separation designs are usually "screened" by air/water tests and verified by steam/water tests. The above limitations usually also imply restrictions of the tests to single separators (or a small number of separators or in some cases scale versions of physically large separators).

Most nuclear steam generators utilize a two-stage system of steam separation, involving both primary separators and secondary separators.

Primary Separators

The primary stage of separation utilizes numerous primary separators in an array supported by a separator support plate (or deck) above the tube bundle. The region between the tube bundle and the separator deck becomes a region for some redistribution of the two-phase mixture exiting from the tube bundle before entrance to individual separators. The separator deck collects the water separated from the steam by the primary separators (which is subsequently routed to the steam generator recirculation loop). It also mechanically supports the primary separators.

In many cases, the flow loading (i.e. steam flow rate and water flow rate) through the primary separators affect the performance of various separator designs differently. In selecting the proper separation equipment for a given application, consideration must be given to the local distribution of two-phase conditions at the entrance to the primary separators and to the water level height which exists on the separator deck to ensure that the equipment selected will operate properly under the predicted conditions.

While there are quite a number of primary separator designs in common use today (and most manufactuters use their own proprietary designs), nearly all use the same principle to achieve separation, i.e. a centrifugal action that utilizes the density differences between steam and water.

Secondary Separators

The secondary stage of separation utilizes numerous separators in an array supported by a plate or structure near the top of the steam generators. The secondary separators are often termed "dryers" because that primarily is their function, i.e. to "dry" the incoming steam/water mixture that rises from the primary separators (mostly steam with some water) via extraction of the moisture to acceptable levels.

In selecting a secondary separation design, consideration must be given to the inlet conditions (i.e. percentage of inlet moisture and inlet velocities) and to the distribution of these conditions in the steam generator. For instance, in U-tube steam generators, there is more steam generation on the hot leg side than on the cold leg. In some cases, the primary separators do not act as an adequate flow distributor. In these cases, added flow resistance (orificing) of the secondary separators has been used to ensure more uniform inlet flow distribution to the various separators in the steam drum.

There are two main types of secondary separators commonly used in nuclear steam generators today. The first utilizes corrugated chevron vanes closely spaced. As the wet steam flows through the "zig-zag" path formed by the chevron elements, water particles adhere to the metal elements and flow downward. The other type uses the same centrifugal principle used in most primary separators.

8.2.4 Steam Generator Thermal/Hydraulics

There are quite a number of thermal/hydraulic aspects of the flow phenomena inside steam generators. Some of these are:

(a) Heat Transfer Area & Tube Bundle Geometry Selection
(b) Hydraulics of the Secondary Side Natural Circulation Loop
(c) Steam Separation Hydraulics and Sizing
(d) Hydraulics of the Primary Side Flow
(e) 100% Power Steady State Thermal/Hydraulic Performance Analysis
(f) Part Load Steady State Thermal/Hydraulic Performance Analysis
(g) Operational Transient Performance Analysis
(h) Abnormal Operation Thermal/Hydraulic Analysis (e.g. Pipe Breaks, Tube Rupture, etc.)

The discussion herein is limited to three of the above, i.e. heat transfer area selection, natural circulation loop hydraulics, and steady state performance analysis.

Heat Transfer Area & Tube Bundle Geometry

The sizing and selection of the tube bundle geometry of steam generators is not a particularly complex matter; however, there are a number of parameters to be selected. These include heating surface, number of vessels per reactor, number of tubes and size, tube pitch and tube length.

In selection of these design parameters, there are a variety of factors involved. Some of these factors impose practical constraints and hence the final selection of tube bundle geometry is usually a compromise. Some of the factors are:

(a) HT steam generator pressure drop inlet.
(b) Tube length limitation due to manufacturing & shipping constraints.
(c) Adequate margin for fouling/plugging/uncertainties.
(d) Reactor building height limitation/drum length.
(e) Tube velocity limits due to material erosion/corrosion considerations.
(f) D_2O inventory.
(g) Manufacturing Economics.
(h) Manufacturing Schedule.
(i) Secondary side thermal/hydraulic requirements.

The surface in the evaporator zone is calculated from elementary heat transfer equations but with a number of interesting refinements.

Surface area A is given by:

$$A = \frac{Q}{U\ T} \qquad (1)$$

The overall heat transfer coefficient U is calculated from:

$$U = \frac{1}{R} = \frac{1}{(r_{hio} + r_{mo} + r_{ho} + r_{do})} \qquad (2)$$

The internal heat transfer coefficient h_i is calculated from the well-known Colburn equation for pipe flow using D_2O flow properties. For the region of D_2O condensation in vertical upflow a forced convection coefficient is usually used, i.e. no heat transfer enhancement is claimed for the condensation behaviour. This probably results in some conservatism in the surface sizing.

The shell side boiling coefficient h_o is predicted by one of the nucleate boiling correlations such as that of (Rohsenow 1952) or (Chen, 1963). The metal resistance is of course determined from the cylindrical conduction equation using the appropriate tube wall conductivity. It is interesting to note that, unlike most heat exchangers, the wall resistance is a large contributor in steam generators to the total resistance (typically 25 to 30%) and hence it is important to accurately predict the wall conductivity of the tubing in the as-built condition. This has required considerable laboratory measurements of tubing conductivity for steam generator tubing materials.

The remaining factor in equation (2) is the fouling resistance, r_d. This factor is an "experience" or "judgement" factor. While very small numbers are used (particularly compared to usual heat exchanger design practice), they have a large effect on the amount of surface required. Normally, this factor is used to provide adequate surface margin not only for fouling or scaling of the heat transfer surface over the lifetime of the unit but also to provide an allowance for tube plugging in the event that this is required. It also provides allowance for other uncertainties such as tolerances on fluid properties and heat transfer correlations.

In equation (1), the overall LMTD in the evaporator zone is not used; a stepwise approach is used that allows the true local saturation temperature seen by various areas of the tube bundle to be employed. For design, steam conditions are usually specified at the steam drum outlet nozzle; hence, to determine the local saturation temperature at any point in the tube bundle, it is necessary to work backwards from the outlet nozzle to the free water surface in the steam drum (accounting for steam separator pressure drop) and then to the particular point in the tube bundle to get the true local saturation pressure and hence temperature. Since there are areas of the tube bundle where the effective temperature differences between D_2O and H_2O are relatively small, this submergence effect is considerable.

The surface in the preheater zone is calculated using fairly conventional techniques that are typical of those for a liquid-liquid heat exchanger. The large physical size of the tube bundle requires some care be taken in applying heat exchanger sizing codes that have been developed for much smaller units. For instance, careful attention must be given to shellside flow leakage paths. In smaller units, tube outer diameter-to-tube baffle plate clearances provide a leakage path that has relatively little effect on thermal performance. However, in typical CANDU S/G preheaters, this leakage path effect is important.

The experience in overall steam generator surface sizing is good. To my knowledge there are no problems relating to the amount of surface in nuclear steam generators except in some PWR steam generators where a very large number of tubes have been plugged due to tubing corrosion. Our recent commissioning feedback on the CANDU 600 plants has provided us with added assurance that the design surface selection techniques are quite reasonable.

Natural Circulation Loop Hydraulics

The recirculation nuclear steam generator (as distinct from once-through units used in some PWR plants) are of the natural circulation type for the secondary side steam/water. The secondary side flow path in a typical recirculating steam generator is comprised of three regimes,, as illustrated in Figure 13. The first is the heat addition regime where heat is added to the shell side secondary water. This consists of a small sub-cooled region and the main boiling region (primarily nucleate boiling). The second regime is in the region above the tube bundle (called the riser) where there is two-phase vertical flow without heat addition. Above the riser is the moisture separation equipment to remove the saturated water from the steam water mixture. The saturated water removed by the moisture separators flows to the top of the downcomer and then flows to the bottom of the downcomer to enter the heating zone and complete the circuit. The natural circulation driving force is provided by the difference between the density of the water in the downcomer and that of the steam water mixture (less dense) in the heating zone and riser.

The driving force for natural circulation flow is resisted by pressure losses which oppose the flow. Calculation of the pressure losses in a steam generator is therefore an integral part of evaluating the circulation and flow rate through the heating zone. In such an analysis (see Figure 14) a quantity known as the recirculation ratio is investigated. The recirculation ratio, R, is defined as:

$$R = \frac{\text{Flow Rate of Recirculated Water in the Downcomer}}{\text{Steam Flow Rate}}$$

A high recirculation ratio is desirable and as mentioned earlier, AECL practice has been to specify a minimum recirculation ratio of 5. The designer can achieve this by selecting a high water level in the steam drum, by minimizing the circuit pressure losses or by some compromise of these. The two-phases losses due to the flow contraction/expansion through the numerous tube supports in the boiling zone and the two-phase loss in the steam separators are normally the two largest losses. Hence, good practice to maximize recirculation ratio is to design hardware that minimizes these losses.

The above consideration of natural circulation loop hydraulics is based on an assumption of steady state conditions. However, there are also considerations of the flow dynamic characteristics of this loop that must be considered in the design.

A temporary change in feedwater flow in a natural circulation steam generator will change the void fraction in the heating zone. This will change the density of the two-phase flow in the heating zone, and the drum water will seek a new level. As a result of feedback in this sequence of events, the water level will start to oscillate, and it is possible for these oscillations to be unstable. Such oscillations are undesirable and should be highly damped because they may create problems in the control of the plant.

Prediction of the dynamic characteristics of a steam generator is a complex problem requiring solution of the time-dependent equations of conservation of mass, energy, and momentum. Solution of such a system of equations is complex and requires the use of a computer. The difficulty of predicting the dynamic behaviour of a steam generator is surmountable because the design parameters that affect water level stability are known. The parameter changes which improve flow stability also tend to accompany an increased recirculation flow rate. Typical measures that may be taken to improve stability include:

1. decreasing the pressure drop in the two-phase zones by increasing the open flow areas in the tube supports, and decreasing steam separation equipment pressure losses;
2. decreasing the void fraction, which is equivalent to decreasing the steam quality, by increasing the recirculation flow rate;
3. increasing the pressure drop in the downcomer section by adding a flow restriction; and
4. increasing the "liquid mass" of the secondary side inventory.

Steady State Performance Analysis

To provide verification of the design, computer analysis of the steam generator thermal/hydraulics is performed. Our analytical predictions are done with a three-dimensional two-phase finite difference computer code developed by AECL. The code, known as THIRST, has the ability to model both primary side and secondary side conditions; handles two-phase homogenous flow; gives flow distributions both on the primary and on the secondary sides; and calculates recirculation ratios.

THIRST provides designers with a tool to analyse steam generator designs, pinpointing, for example, areas of flow stagnation in the secondary side, where corrosion would be enhanced and heat transfer inadequate. Appropriate design changes can then be made to correct the flow patterns. Designers can also examine local phenomena which determine overall efficiency, and then maximize the performance of the unit through an iterative process by altering design features.

In modelling the steam generator tube bundle, the complete region from the top of the tubesheet to the top of the U-bends is included. This region is typically divided into approximately 5000 control elements, with concentration of the elements near the tubesheet area.

Input data required for the analysis of a steam generator involve the geometric layout; primary fluid inlet enthalpy, flow and pressure; secondary feedwater enthalpy and flow, outlet steam pressure, and normal operating water level. Eight different steam generator designs have been analysed. These have included a number of diverse features such as an integral preheater, square and round U-bends, and a range of feedwater entrance geometries.

Velocity and quality distribution resulting from a typical THIRST analysis of the design discussed above are shown in Figure 15 and 16. Quality values are marked directly on the contours in Figure 15. The preheater section is below saturation, except for a small zone at the exit. The conical shape of the contours on the hot side illustrates the penetration of the downcomer flow into the center of the tube bundle. The lower horizontal cut shows the quality pattern on the tubesheet face. Just above the preheater, the mixing of the higher quality fluid from the hot side with saturated liquid from the preheater results in steep gradients of quality on the cold side. As the mixture moves up through the remaining bundle the quality continues to rise. The hot side qualities are higher throughout.

The influence of the U-bend geometry can be seen by the shape of the quality patterns at the top of the vertical plane. This is more obvious from the velocity vectors in Figure 16. In the U-bend region, the fluid tends to migrate out towards the shroud where the resistance to flow is lower. Just below the U-bends, the velocities are primarily axial and parallel to the tube bundle. At the preheater exit the higher flow on the hot side redistributes over the cold side. In the preheater the zig-zag pattern of the flow around the baffles is somewhat difficult to see in Figure 16 because these velocities are relatively small. At the tubesheet face the downcomer flow is shown as coming in on the hot and cold side and converging near the center of the hot side. This corresponds to the point of highest quality near the tubesheet in Figure 15.

Details of this computer code and some examples of the type of parametric studies that can be done are available (Inch, Shill, 1980) for the interested reader.

8.3 PHT Pumps

8.3.1 Single Phase Performance

To establish the design point of the PHT pumps at the full power condition, hydraulic calculations for the heat transport system are done to obtain the most probable circuit pressure drop between the pump suction and the pump discharge. A small margin is then usually added to the calculated pump head requirement to allow for uncertainties, such as the two-phase pressure drop calculation in the circuit. This procedure establishes the design head needed from the PHT pumps at the design flow in the loop. The pump design is optimized for peak efficiency at these conditions.

In CANDU reactors, the pressure tube and fuel design characteristics lead to higher head, lower flow pumps compared with those of the PWR's (see Table 5). Hence, the specific speed characteristic number for CANDU pumps allow use of a volute type casing. The higher specific speed of PWR primary system pumps requires the use of spherical (onion shaped) casing with internal diffusers around the impeller (see Figure 17). A benefit of the volute type casing for CANDU systems is its low hold-up volume of expensive heavy water.

The detailed hydraulic design of the pump casing and impeller are set by the pump manufacturer and to some extent each manufacturer considers these details proprietary. However, basic parameters are quite well known. Impellers are single sided and shrouded. From a mechanical design standpoint, the fully shrouded impeller is desirable as it reduces the axial thrust compared to the open impeller design. The volute casing design (or scroll) has advantages in their simplicity and relatively low cost. The volute of the pump is generally designed to be a collector only (i.e. the increase in area is such as to accomodate the increments in flow from various impeller passages). However, an important parameter in the design of the pump, after the hydraulics have been settled, is the side load (radial thrust) on the shaft produced by the impeller since it influences bearing design. Theoretically, if the velocity is constant in the volute, then the static pressure is likewise constant around the impeller circumference and the radial thrust will be zero. At best, the radial thrust can only be reduced to zero (or a minimum value) at the design capacity, since the volute area will not be correct for other flows. The objective is then to ensure that bearings are large enough for the range of off-design conditions.

There are several areas of pump technology where state-of-the-art analytical techniques are not sufficient to adequately verify hydraulic and mechanical performance. This leads to requirements for testing of each completed pump/motor unit at the manufacturer's plant. These tests include tests at off-design conditions to determine the characteristics of the assembly over the whole range of flow from near-zero to flows typically up to 140 to 150 percent of design flow. From the tests the characteristic curve is established for each pump giving actual data on head pump power and efficiency versus flow (see Figure 18 for a typical example). As well as performance tests, endurance and startup/rundown tests are performed.

To test the pumps and motors thoroughly under simulated reactor conditions and particularly off-design conditions is important. This is because, in the plant, the pumps may run at a point other than design on their performance curve (at least for a while). For instance, if reactor power or other conditions (e.g. steam generator surface cleanliness) are such that two-phase conditions are lower than expected, then the operating pump flow will be higher than the design flow. Another example is during commissioning when there is no fuel in the reactor, flows in the heat transport system are considerably higher than the normal design values.

There are two other aspects of pump performance worth noting. The first is prevention of cavitation which is an important phenomenon in centrifugal pumps. It is essential to assure sufficient net positive suction head (NPSH) to prevent pump cavitation during the various expected operating modes of the heat transport system. This equipment restriction imposes limitations on the system designer on conditions when there is low pressure (and hence low NPSH) in the heat transport system, e.g. system warmup and events leading to system depressurization during operation. Minimum NPSH requirements of the pumps are established by testing.

The other aspect is the large pumping energy transferred to the coolant. While this source of heat is relatively insignificant in the design of most systems, (and even here the pump energy is less than 1% of the fission energy) the absolute value is significant, i.e. typically 17 MW(th) in a CANDU 600 reactor. Hence it is accounted for in the heat transport system heat balance.

8.3.2 Design Features

CANDU heat transport pumps are vertical, centrifugal, single stage, volute machines driven by an overhead directly-coupled, electric induction motor (see Figure 19). This configuration was chosen originally because it was the optimum size and shape for the space available in the reactor building. The single shaft seal assembly and the external thrust bearing in the overhung pump is preferred to the double seal assembly and extra guide bearings which one finds in horizontal double shaft pumps.

The shaft sealed pump design dominates in CANDU and PWR utility reactor heat transport pumps. None of the various canned pumps (in which the motor is hermetically sealed and hence features zero-leakage) has gained acceptance in large reactor plants, because of the efficiency penalty and because the very large pumps required, or alternatively the large number of small pumps required, present significant design problems. The shaft seals are considered in greater detail in the next section.

Next, a few words about the number of pumps from the equipment design perspective. Initially, reliability was sought by using smaller pumps and including extra pumps as spares in the circuit (the smaller pumps were considered more reliable because of the smaller shaft seal size). Thus, Pickering has 16 pumps per reactor with only 12 needed to achieve full power. On later reactors, 4 pumps were selected with no installed spare pumps (see Table 6). Fewer pumps lend themselves to better monitoring and maintenance. In addition, the system is simpler to control and there is no need for valves with their associated unreliability. In general, the Bruce system with fewer, larger pumps has proved just as reliable as the Pickering system with its many pumps. Further, the simple arrangement may yield a higher ultimate reliability. Therefore, the four pump arrangement is used on the CANDU 600 and it has been chosen for the CANDU 950 (Brooks, 1981).

As shown in Figure 19, each pump is driven by a large vertical, totally enclosed, air-water cooled, squirrel cage, induction motor. A removable shaft coupling connects the motor to the pump. Removal of the coupling allows sufficient space for the pump seals to be removed without removing the motor.

The pump/motor rotating assembly design must provide sufficient rotational inertia to prolong pump rundown after loss of power. Physically, the majority of this inertia must be provided in the motor portion of the set and earlier CANDU heat transport motors were equipped with large flywheels. In later designs, the forged flywheel was eliminated in favour of packages of laminations, similar to motor rotor punchings, located on each side of the rotor core. This eliminates the need for in-service flywheel inspection and provides a better balanced motor rotor assembly.

Each pump-motor set is provided with a brake capable of stopping an unpowered pump subjected to reduced flow conditions.

The pump and motor rotors are directly coupled to form a common rotor system (Figure 20) that is supported by several bearings. These typically are a tilting pad guide bearing at the top and bottom of the motor, a double acting tilting pad thrust bearing at the top of the motor and a guide bearing (hydrostatic or hydrodynamic) in the pump. To allow for slight guide bearing misalignment between the pump and motor, the motor half coupling has a small amount of flexibility built into it.

The motor bearings are lubricated by cooled and pressurized oil. An oil "lift" system supplies high pressure oil to both sides of the thrust bearing simultaneously during start-up to avoid wear during starting and stopping due to an inadequate oil film. Tilting pad bearings were chosen in part because they generate a film of oil which avoids rubbing contact and gives promise of a bearing lifetime that is at least equal to the station lifetime (30 to 40 years).

Tilting pad bearings are particularly suitable for a vertical machine because each pad (of an eight pad arrangement) produces its own oil wedge and thus provides a centering force on the shaft. Considerable detailed design attention is given to the engineering of the bearing installation and to ensuring an adequate supply of clean oil.

TABLE 5 - COMPARISON OF TYPICAL CANDU AND PWR PUMPS IN LOOPS OF EQUAL POWER

	CANDU	PWR
Power per loop	300 MW	300 MW
Pumps per loop	2	1
Design Pressure	1870 PSI	2250 PSI
Design Temperature	535°F	550°F
Head	705 ft.	300 ft
Flow	29400 IGPM	71000 IGPM
Power (Hot)	7000 HP	5700 HP
Speed	1800 RPM	1200 RPM
Specific Speed*	2470	4860
Impeller Type	Centrifugal	Mixed Flow
Casing	Volute type	Spherical (Onion shaped) with diffusers
Motor	Squirrel cage induction	Squirrel cage induction
Pump/Motor configuration	Vertical	Vertical

$$* \frac{\text{RPM (US GPM)}^{\frac{1}{2}}}{\text{Feet}^{3/4}}$$

TABLE 6 - CHANGES IN PUMP NUMBER AND SIZE

Station or Unit Type	Installed Pumps Pumps Per Reactor	Pumps Needed For Full Power	Pump Power MW (cold system)
NPD	3	2	0.60
DOUGLAS POINT	10	8	0.93
PICKERING	16	12	1.40
BRUCE	4	4	8.20
CANDU 600	4	4	6.70
CANDU 950	4	4	12.00

The combination of large shaft and large bearing sizes together with the type of bearings chosen have resulted in smooth running machines with very low vibration levels.

The detailed design and choice of the pump bearing has traditionally been selected by the pump manufacturer. The choice is between the hydrostatic and the hydrodynamic type. Both bearing types have been used. Both types are lubricated by heavy water. The hydrodynamic bearing requires a guaranteed supply of cool heavy water whereas the hydrostatic bearing can be lubricated by heavy water at HT system operating temperatures, if needed.

8.3.3 Seals

Seals have traditionally been the shortest lived, most unreliable component of the heat transport pumps, as in most other nuclear and non-nuclear pumps. They have accounted for the major portion of the Pickering pump unreliability. A similar situation existed in the PWR program where a mean seal lifetime of 1.3 years was reported from a survey of 19 nuclear stations in the U.S. from 1972 through 1975. After early problems, the mean lifetime of primary pump seals replaced since 1973 in the Pickering station has been 3.3 years. For newer CANDU stations, a target of 5 years shows signs of being achieved from the start with a Canadian-modified seal design (Metcalfe, 1979).

Increases in reliability have been achieved by improvement in seal design, quality assurance, operating policy and feedback from detailed examination of seals after service.

Seal Design

The typical arrangement of shaft seals consist of two (or three) identical high pressure mechanical seals in series and a backup seal. These mechanical seals have a rotor and a stator in close contact. They are described as a balanced, face type seal (or radial flow seal) in which minimal heavy water leakage between the faces provides a fluid film for lubrication purposes. Although two (or three) seals are used in series, each one is capable of withstanding full system pressure in the event of failure of the other(s). This feature has proved essential in avoiding forced shutdowns. The internal pressure which is monitored, provides timely warning when replacement is necessary.

Seal design has been a matter of continuing concern by AECL and significant advances have been made in understanding the factors involved in long seal life. Considerable development at AECL's Chalk River Nuclear Laboratories (CRNL) has provided methods to calculate and control seal component deformations due to pressure and thermal effects using sophisticated finite element computation techniques. For instance, face and holder cross-sectional profiles have been modified until shapes were found (see Figure 21) which would rotate to a slightly open position at the outside diameter upon pressure rise, thus providing greater leakage and lubrication. Similarly, seal face profiles can be chosen which respond to temperature increase by deflecting slightly open, allowing the fluid film to re-establish itself.

Seals that were designed using this technology are performing successfully in the Bruce units and in a number of CANDU 600 plants.

Seal Quality Assurance

Due to the complexity of manufacture and preparation, the seal package quality is very subject to human error. Therefore, special quality assurance provisions are now used and these have been very effective in improving seal reliability.

Quality assurance starts at the factory and continues through the seal assembly and installation phases. For instance, throughout manufacture, controls are placed on the various heat treatment, machining, lapping and inspection processes. Each piece is identified with a serial number allowing a record to be kept in an accompanying Q.A. docket of the operations and measurements associated with that piece. Parts are checked just before assembly for critical flatness and roundness characteristics and the package is assembled only just before it is needed so that storage deterioration is eliminated. Pressure test checks are also made of each seal package just prior to installation in the pump.

Seal Operating Policy

At Bruce, a policy has been used of replacing seals on a planned basis, during reactor shutdowns, when the seals get close to their estimated life expectancy, even though they may still be functioning well. In addition, seals which give abnormal readings (e.g. high interseal pressure) are replaced as well. This operating policy has prevented forced shutdowns due to seals.

Seal Examination after Service

Utilizing procedures developed during laboratory testing of seals, the Bruce seals removed from service are subjected to detailed examination by CRNL. The examinations have already been instrumental in identifying incipient failure mechanisms and the means to avoid them. The objective of a reliable 5 year seal life is within sight.

For the interested reader, further details are given in (Earl, 1983).

8.3.4 Abnormal Operation (Earl, 1983)

Certain abnormal conditions are postulated that affect the heat transport pump operating conditions; capability of the pumps to operate under these abnormal conditions is investigated. Two such events considered below are: loss-of-service-water and a rupture of the heat transport system (LOCA). The pumps are not expected to endure these abnormal events within the design limits for normal operation. For instance, damage is acceptable as long as it does not stop the machine from operating for limited periods of time in the abnormal condition.

Loss of Service Water

Service water is required for cooling the pump gland, the motor bearings and the motor stator winding. Tolerance to loss of service water is built into the design by:

- Providing two mutually redundant gland cooling systems - if service water fails, the gland is still cooled by the heavy water gland injection system.

- The low operating temperature and large thermal capacity of the motor stator so that the insulation will last and the motor will run for at least an hour without stator cooling.

- Using large oil reservoirs for the motor bearings, which are then slow to heat-up and will run for at least an hour without service water.

- Using a 'hydrostatic pump' bearing which is energized with hot heavy water from the pump volute case and does not require cooling.

The ability of the pump-set to continue running without service water for limited times has been demonstrated by actual tests with the pump operating in a test loop. Also, analysis of the pump-set has been performed.

LOCA

In the event of a LOCA, the pumps would be kept running to provide forced circulation over the fuel and consequently greater assurance against fuel overheating and the resulting economic losses. The conditions (pressure temperature, void) at the pump suction vary depending upon the size and location of the LOCA and the time into the event. They range from sub-cooled liquid to a fluid with a 90 per cent void fraction.

It is a well known phenomenon that pumps operating with insufficient suction head exhibit considerable vibration. The performance of CANDU heat transport pumps under these conditions has been evaluated by testing.

The pump testing, performed by Ontario Hydro in their Dobson Research Laboratories, has included the range of suction conditions predicted from the thermohydraulic analyses of the heat transport system. The hydraulic performance of the pump varies from full pump head to a degraded head which is close to the normal head of the pump expressed in height of vapour. The highest vibrations were experienced during head degradation at lower temperatures (Figure 22). To produce void in the loop, a known volume of liquid was drained during operation.

Following testing, the pump and motor were dismantled and inspected. Although the pump bearing showed signs of rubbing, it was still in a serviceable condition. The testing has provided valuable information on the hydraulic performance of the pump and has demonstrated the mechanical integrity of the pumps under suction void conditions that would be experienced in a LOCA event.

Hence, a variety of measures, such as actual tests, design features and/or analysis, are used (depending on the event considered) to demonstrate the capability of the pumps to endure abnormal operation conditions.

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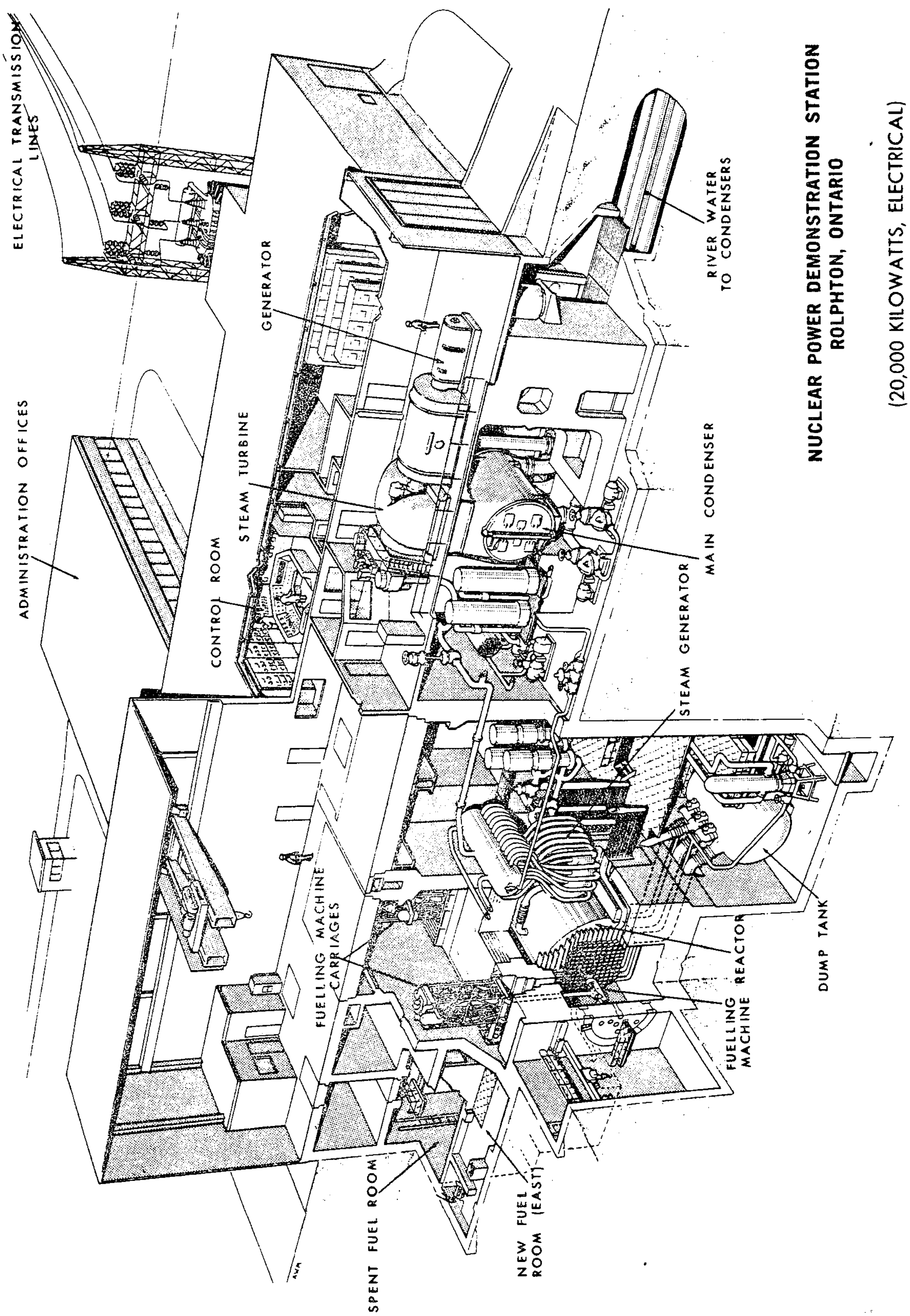


FIGURE 1 CUTAWAY VIEW OF THE NUCLEAR POWER DEMONSTRATION PLANT

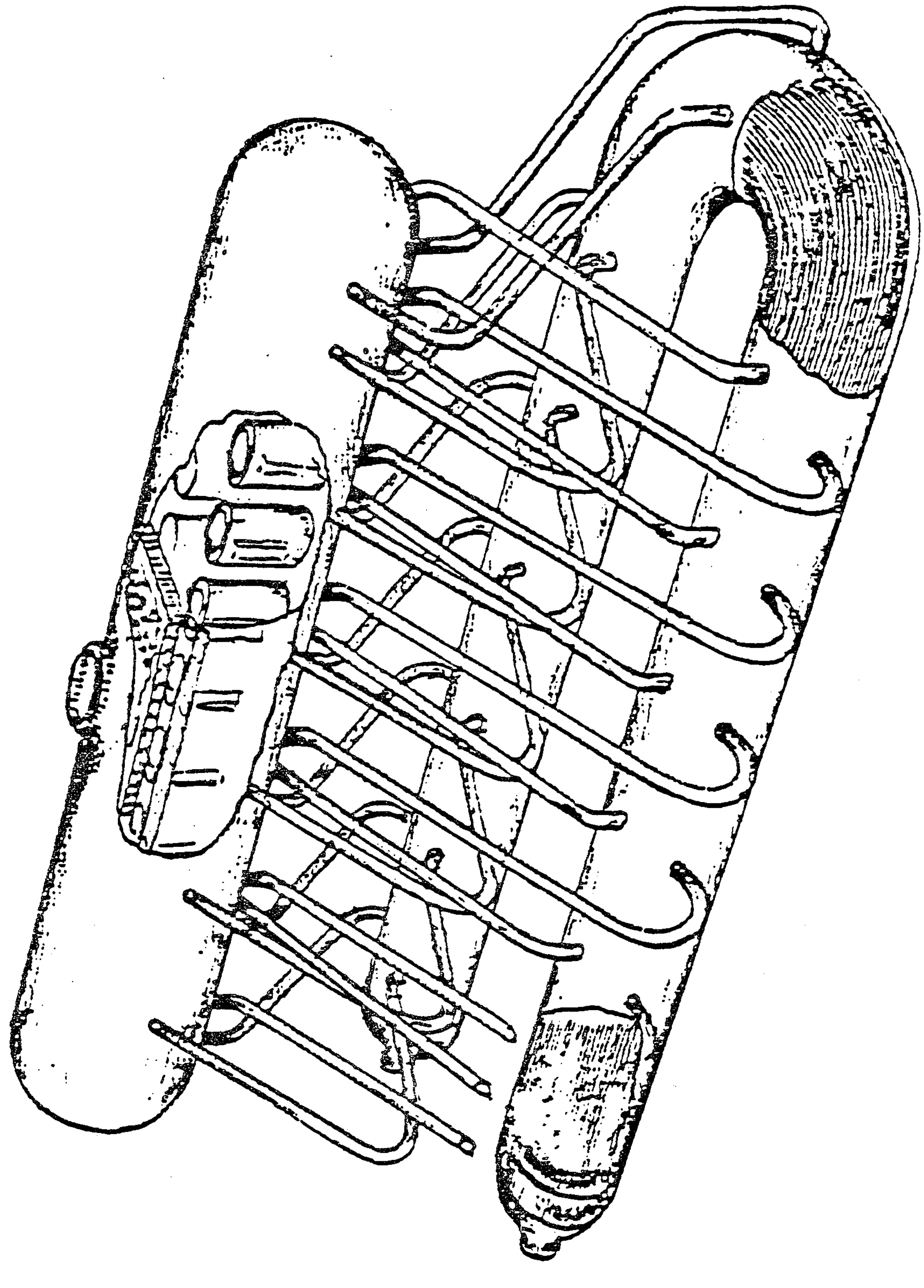


FIGURE 2 SHIPPING PLANT HORIZONTAL TUBE STEAM GENERATOR

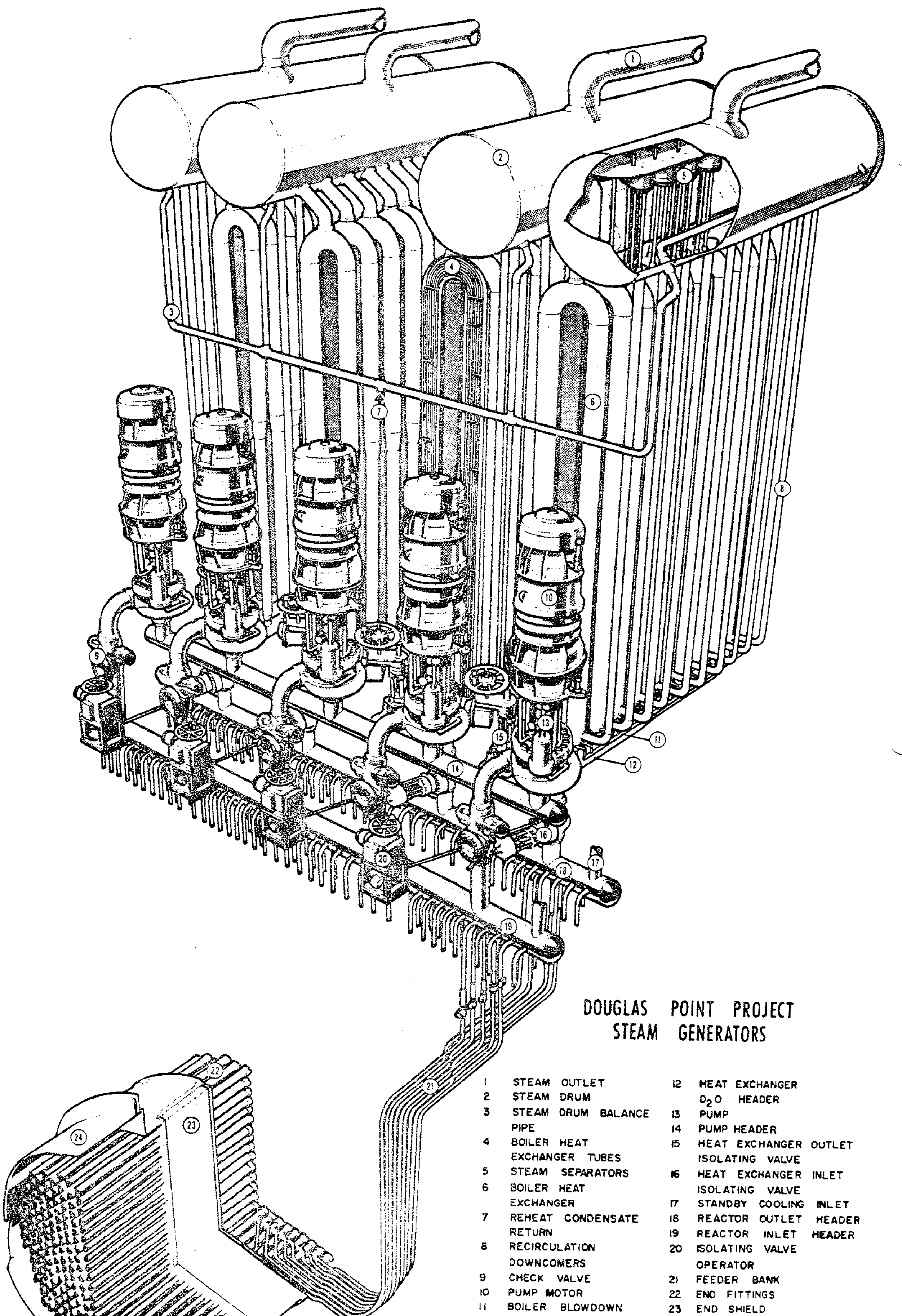


FIGURE 3 DOUGLAS POINT MAIN COOLANT CIRCUIT EQUIPMENT

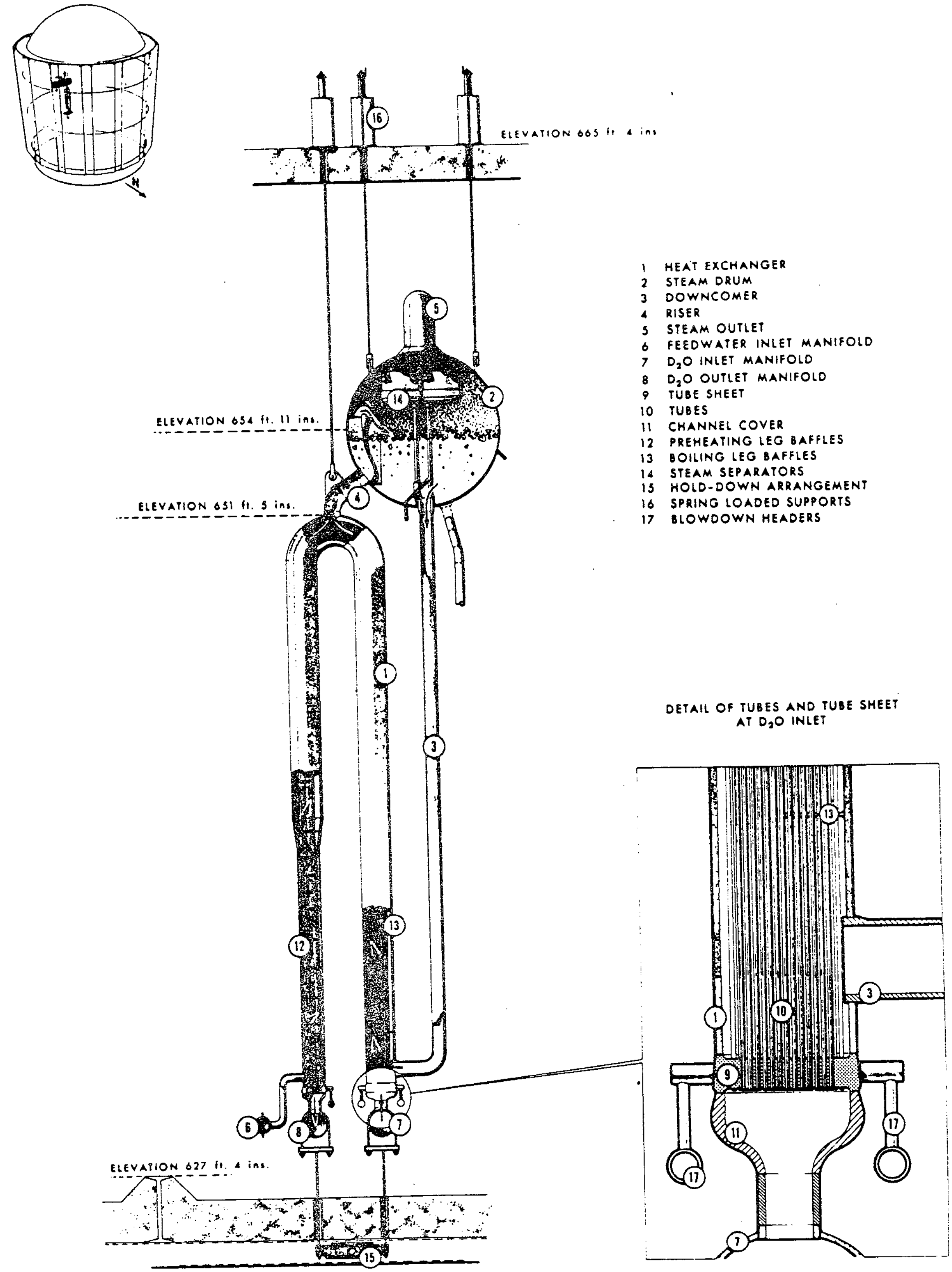


FIGURE 4 DOUGLAS POINT PRIMARY SYSTEM STEAM GENERATOR

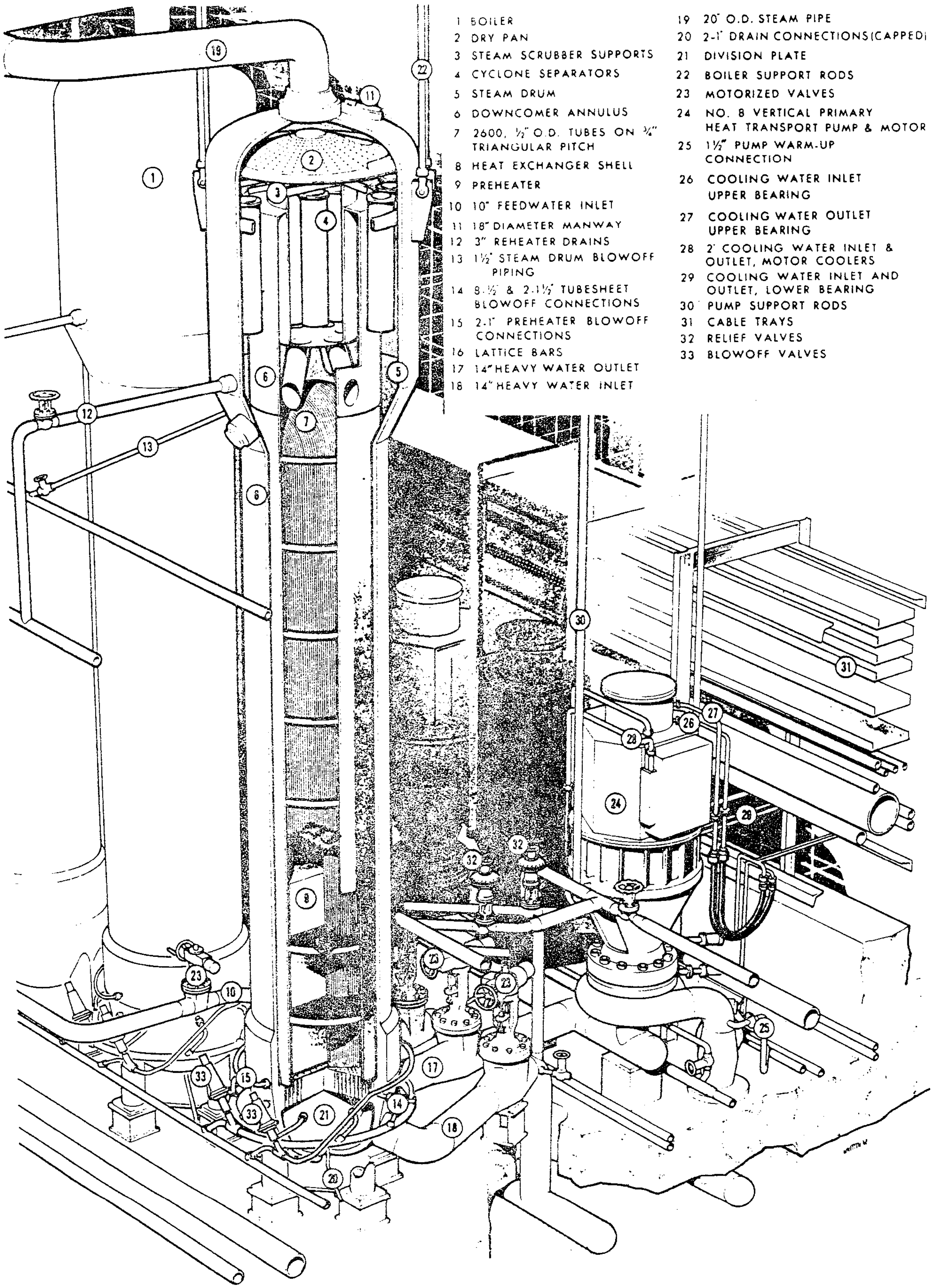


FIGURE 5 PICKERING 'A' PRIMARY SYSTEM STEAM GENERATOR AND PUMP

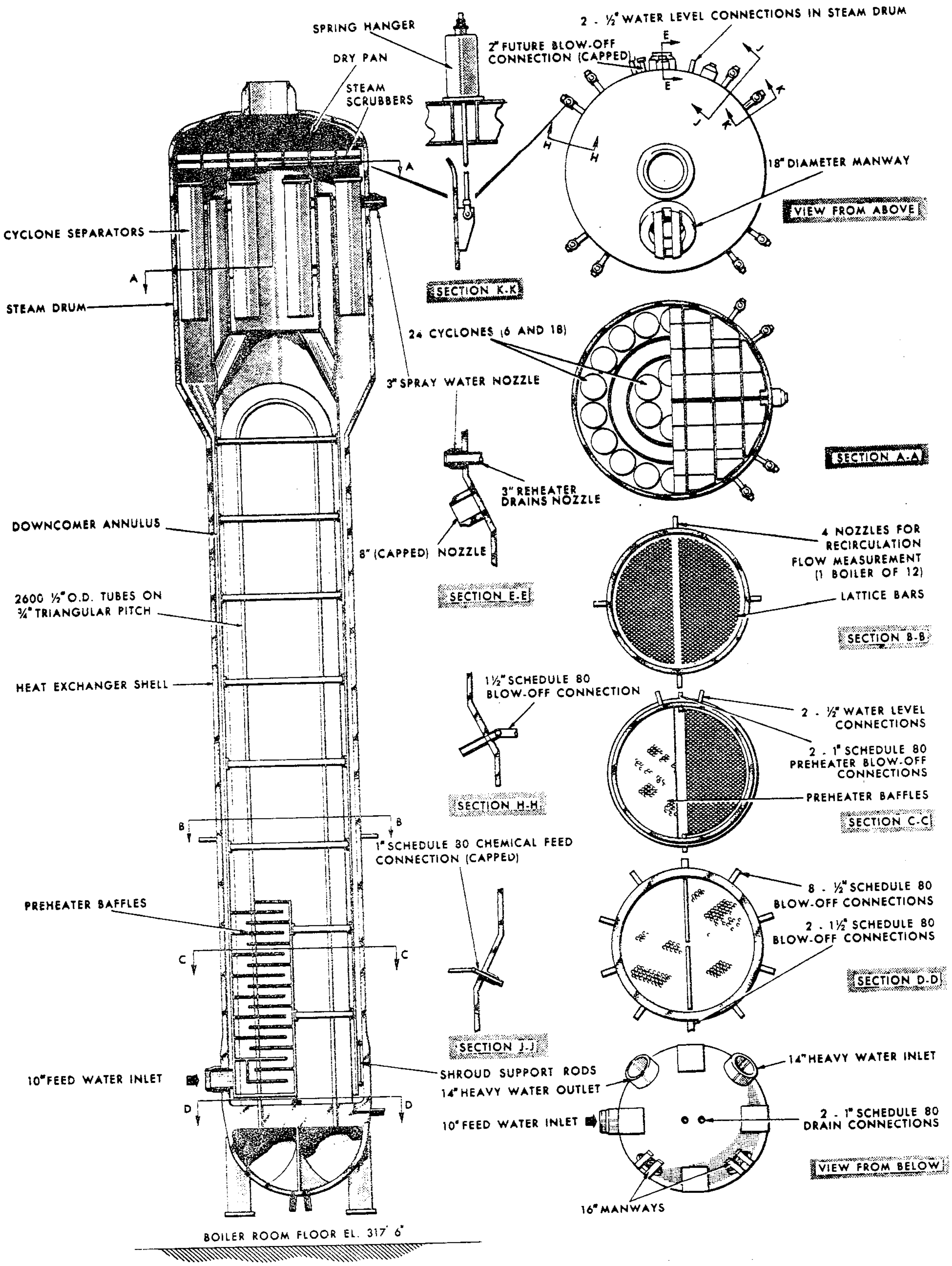


FIGURE 6 PICKERING 'A' STEAM GENERATOR SECTION

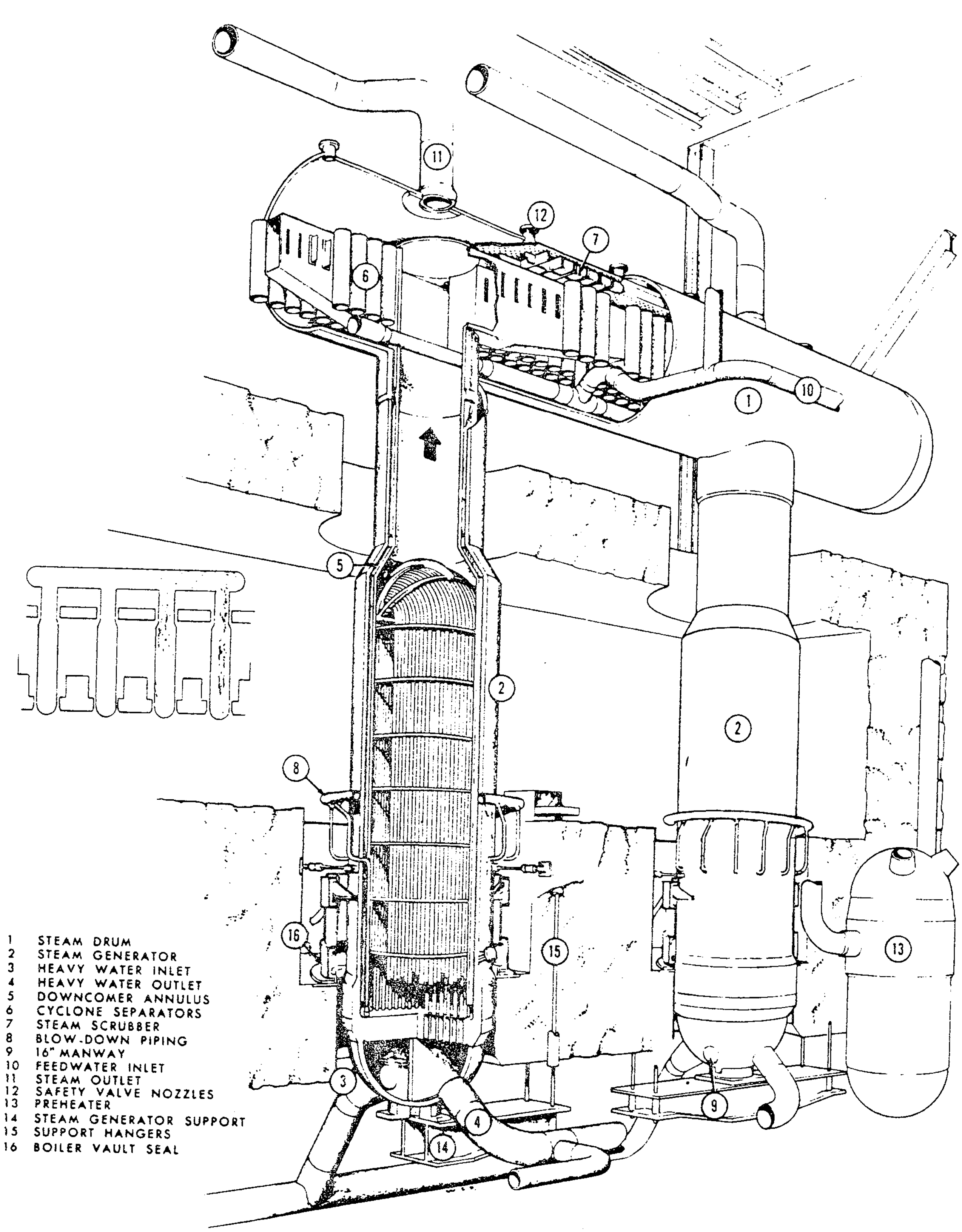


FIGURE 7 BRUCE 'A' STEAM GENERATOR

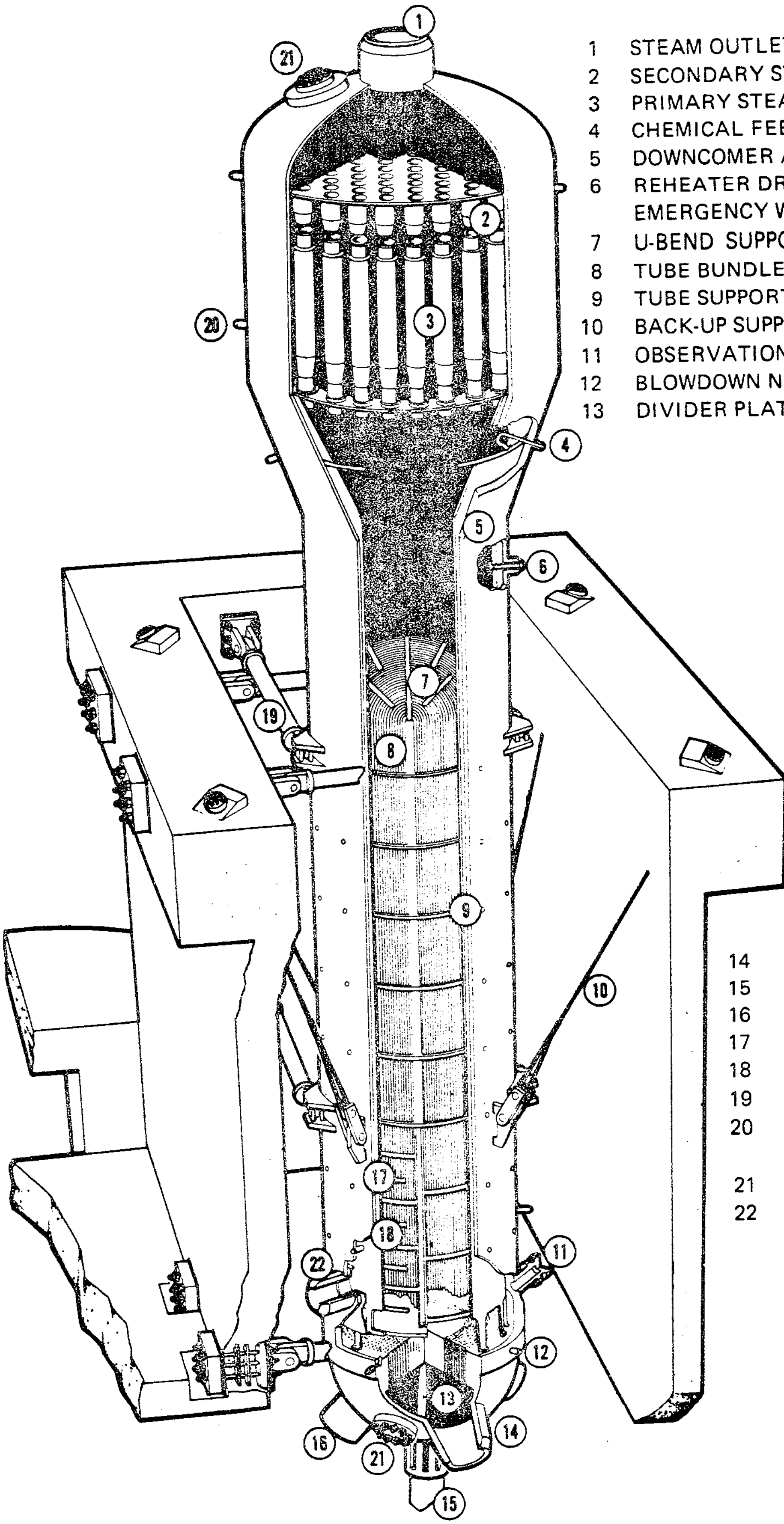


FIGURE 8 600 MW(e) STEAM GENERATOR

NUCLEAR STEAM GENERATOR

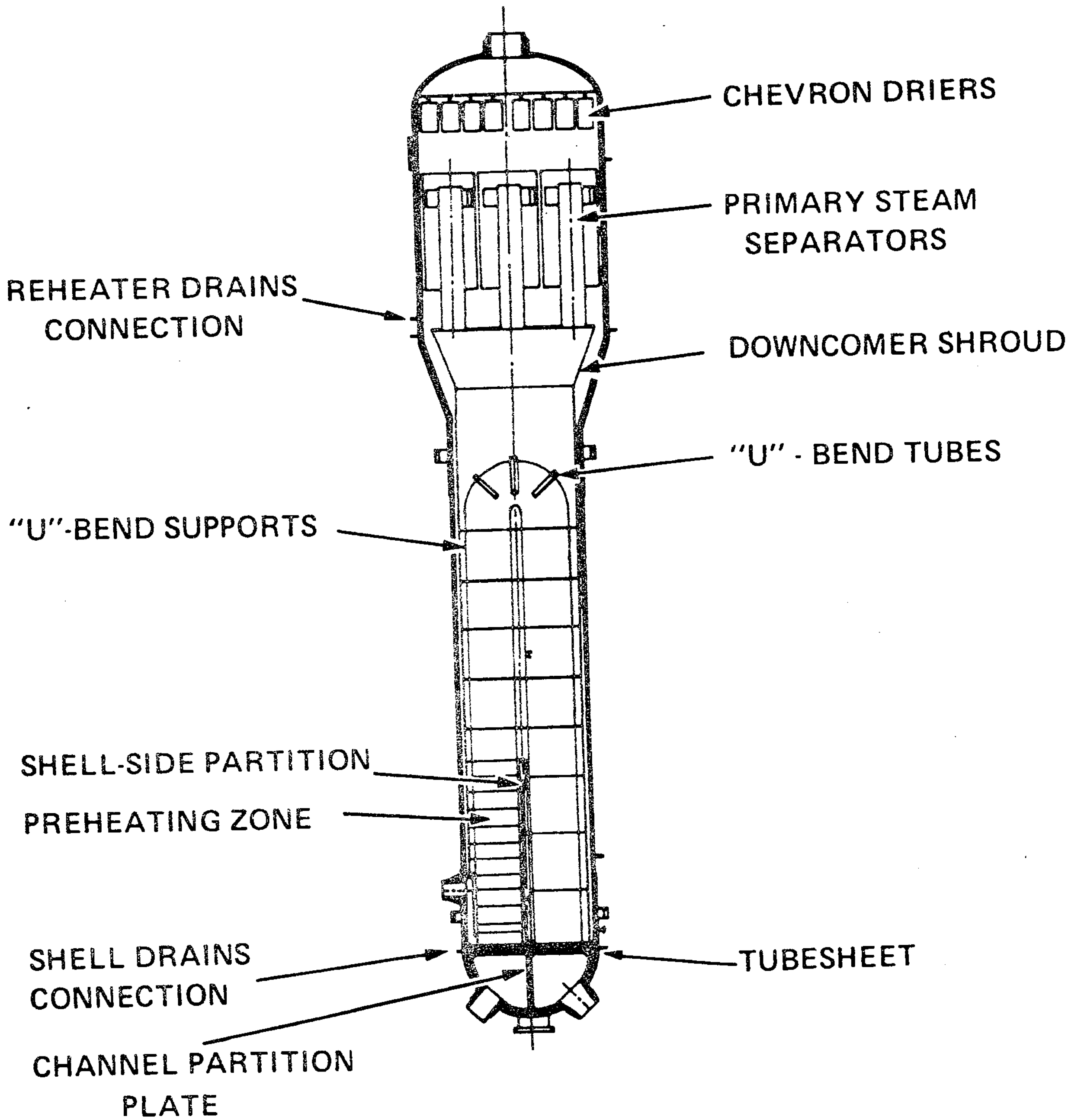
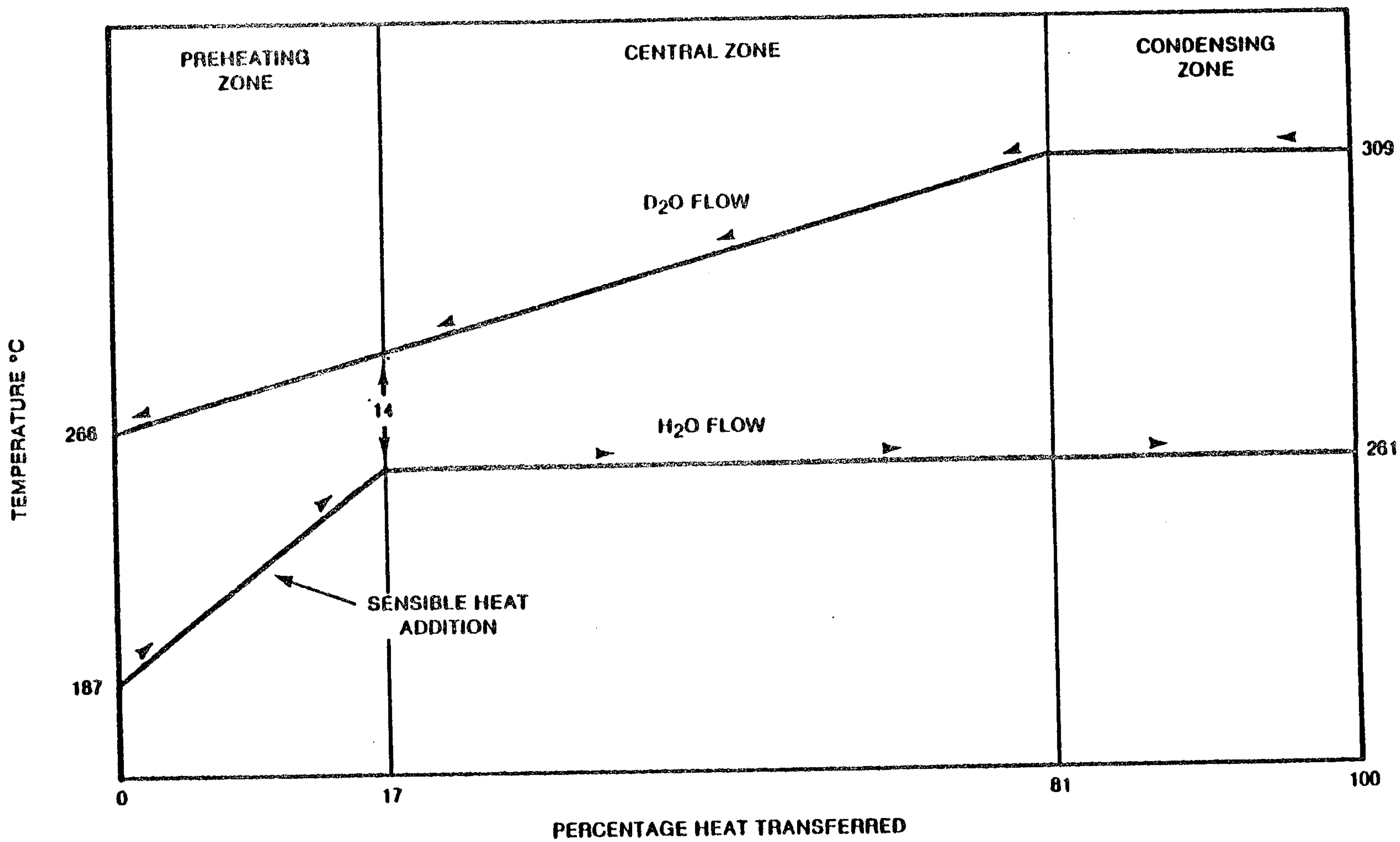


FIGURE 9 CROSS SECTION OF 600 MW(e) STEAM GENERATOR



Data presented is for AECL 600 MW(e) reactor

FIGURE 10 STEAM GENERATOR HEAT LOAD DIAGRAM

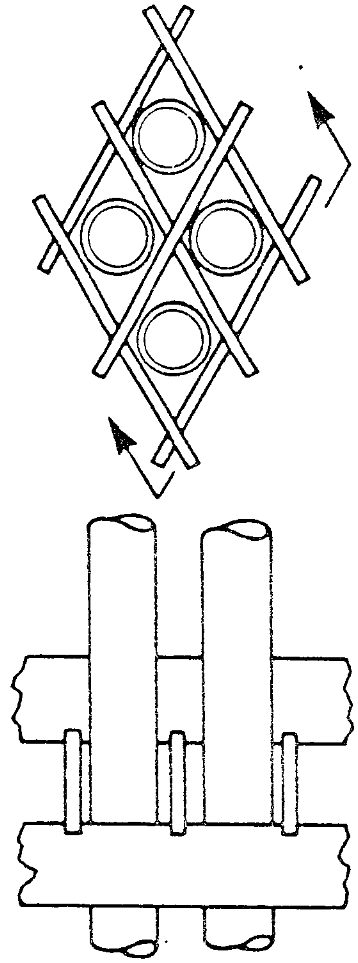


FIGURE 11 LATTICE BAR TUBE SUPPORTS (PICKERING A S.G.)

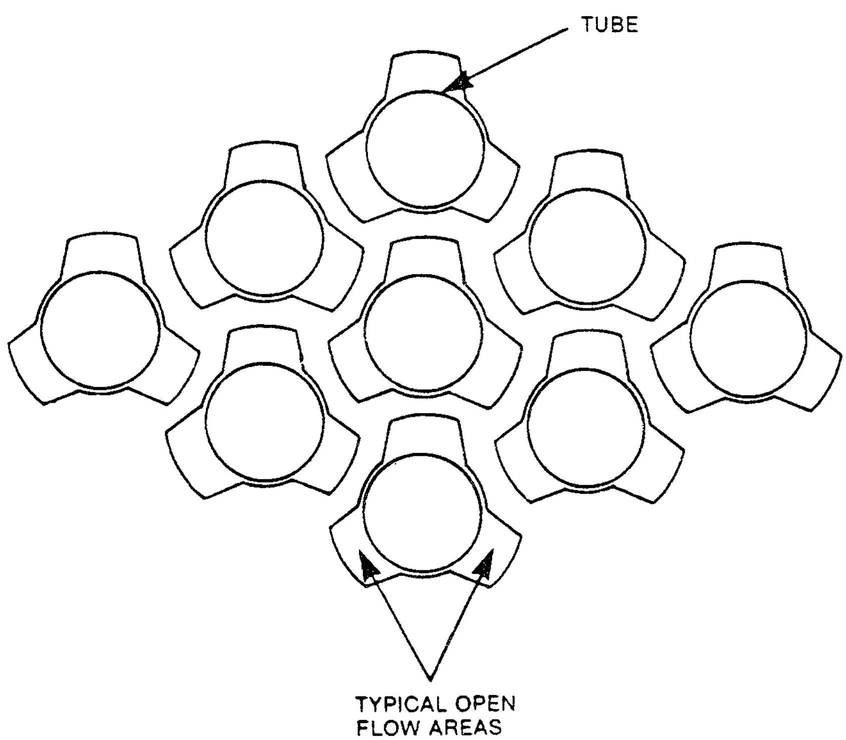


FIGURE 12 TUBE SUPPORT PLATE USING TRILOBAR BROACHED TUBE HOLES

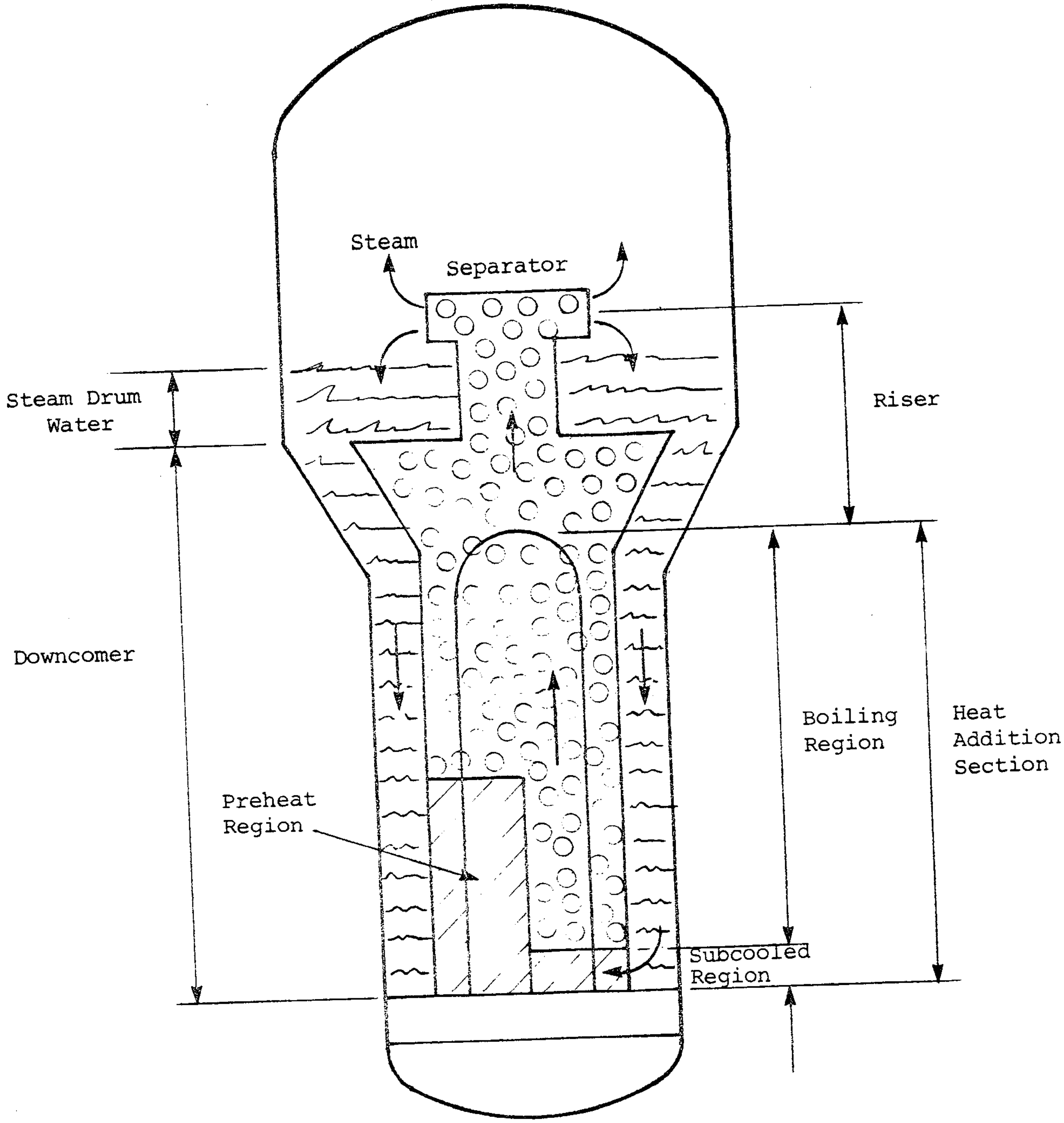


FIGURE 13

FLOW REGIMES OF TYPICAL
RECIRCULATING STEAM GENERATOR

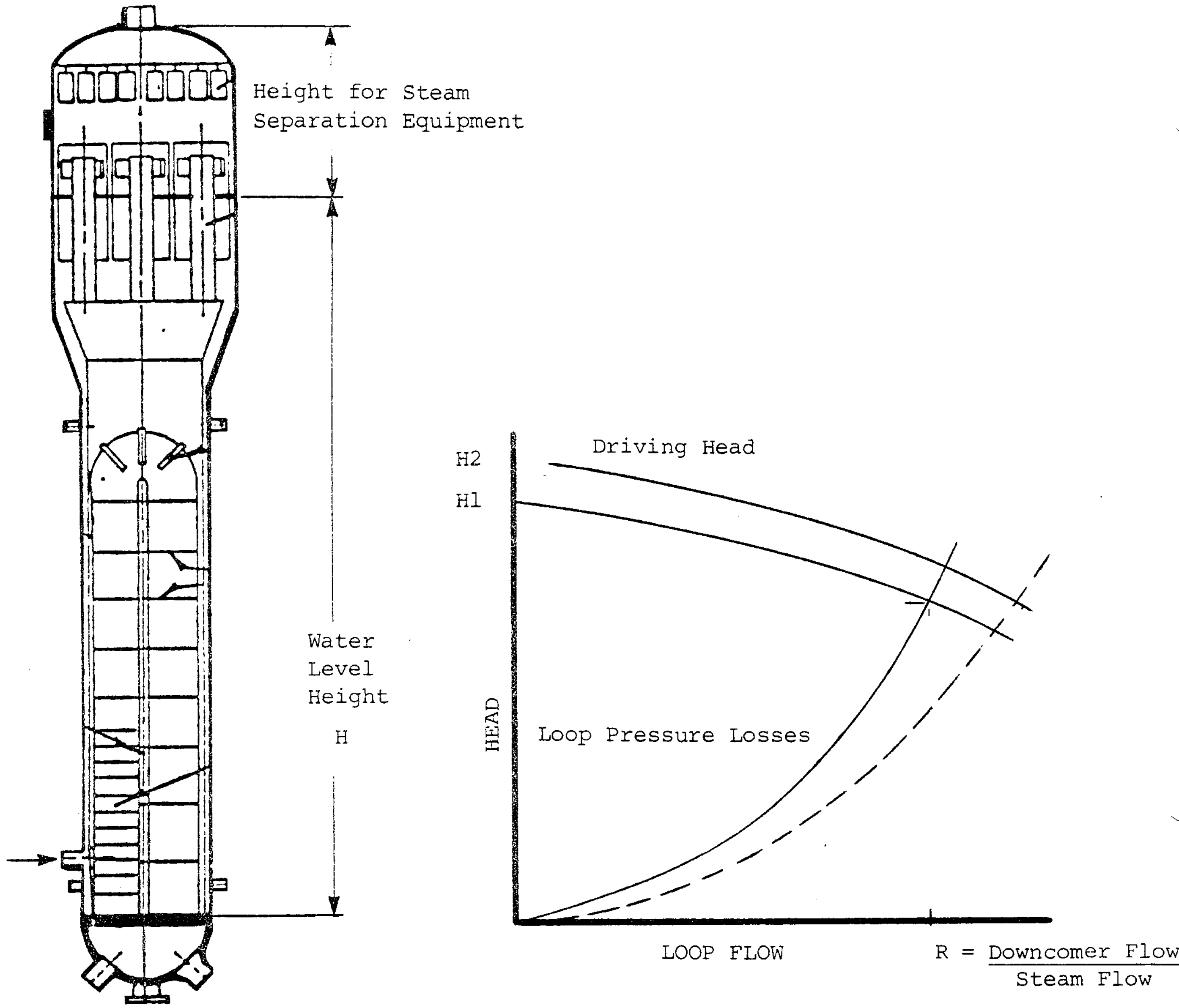


FIGURE 14

CANDU S/G SECONDARY SIDE NATURAL CIRCULATION LOOP CONSIDERATIONS

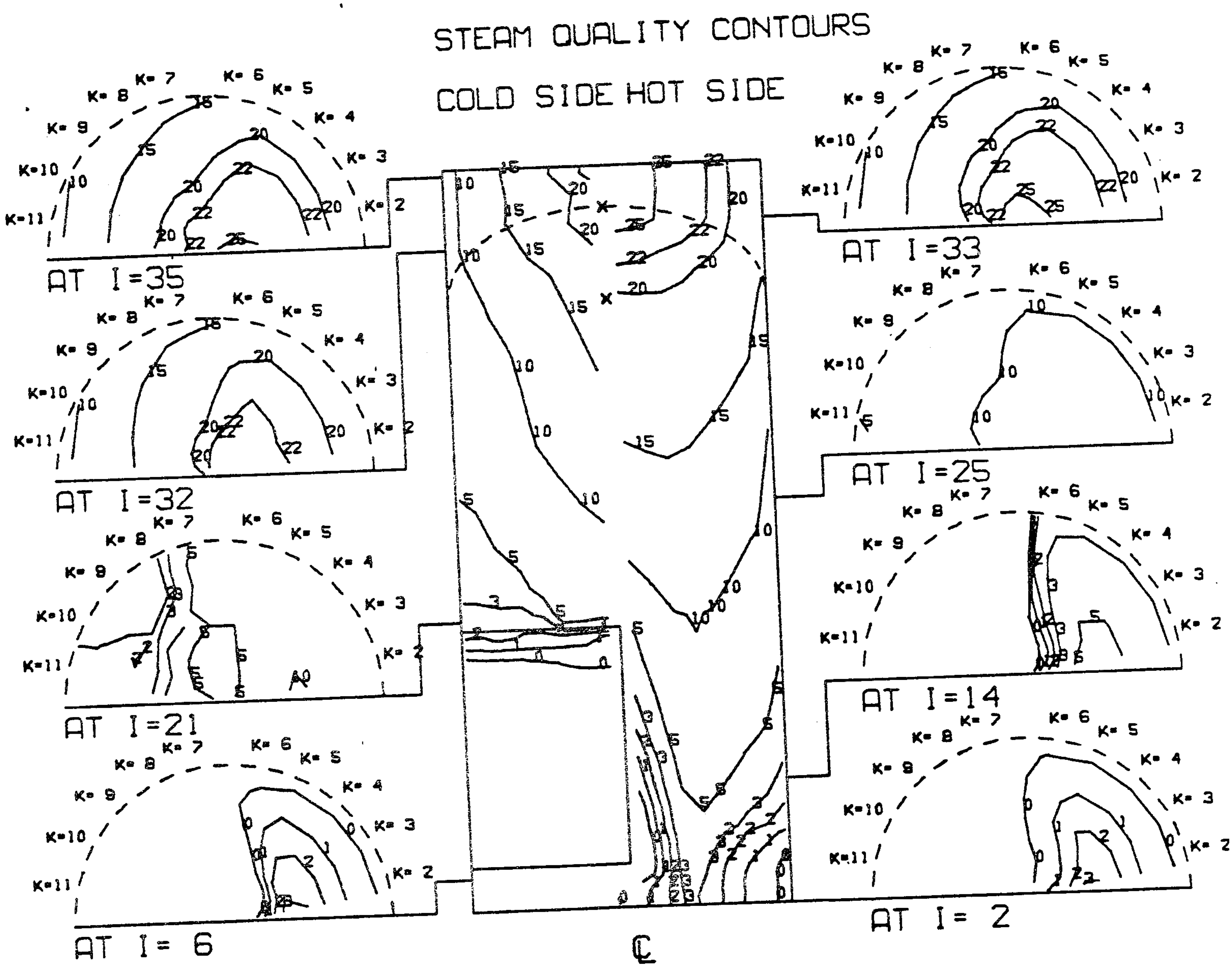


Figure 15 THIRST OUTPUT - Composite Plots (Quality Distribution)

Taken from Inch, Shill 1980

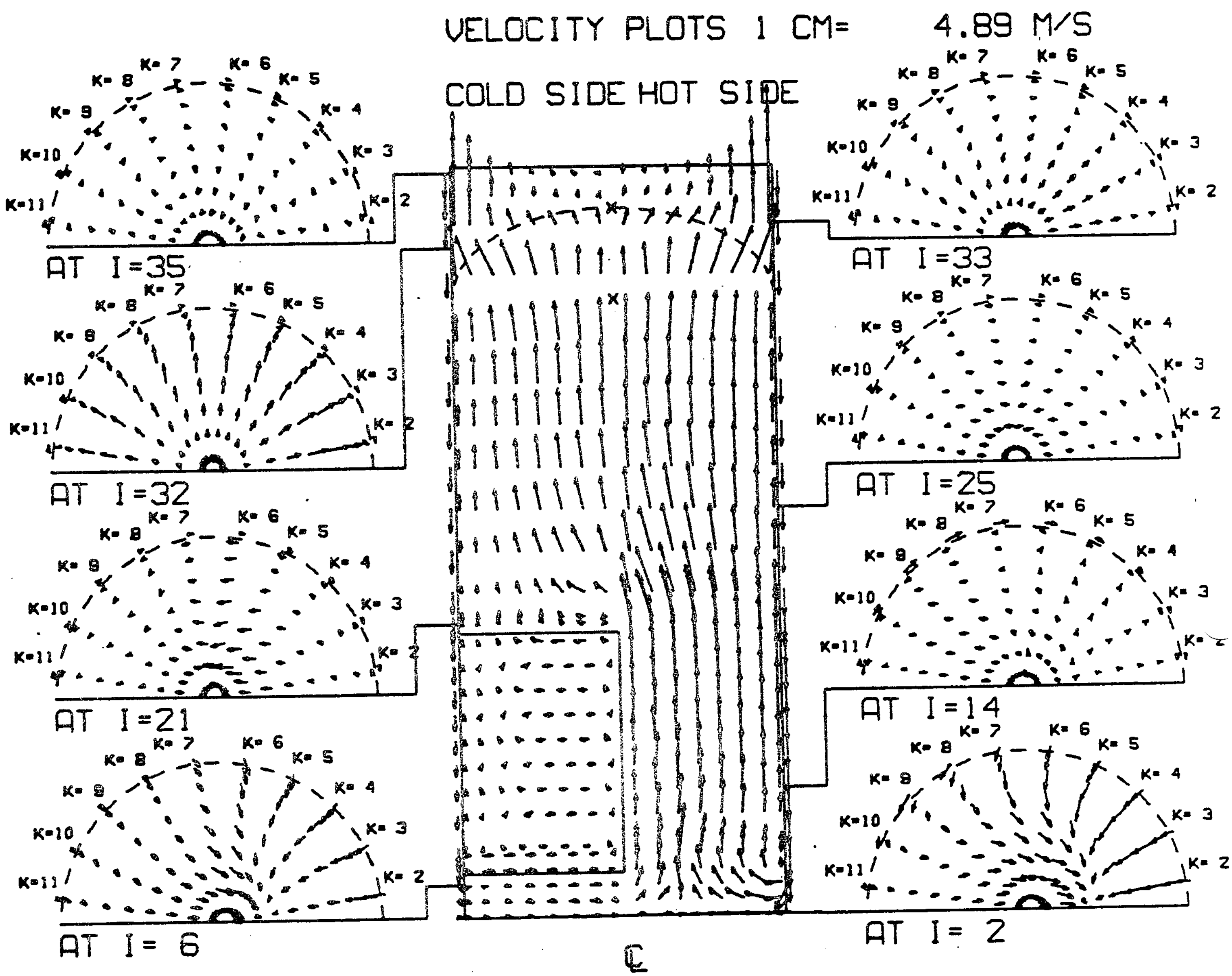


Figure 16 THIRST OUTPUT - Composite Plots (Velocity Distribution)

Taken from Inch, Shill 1980

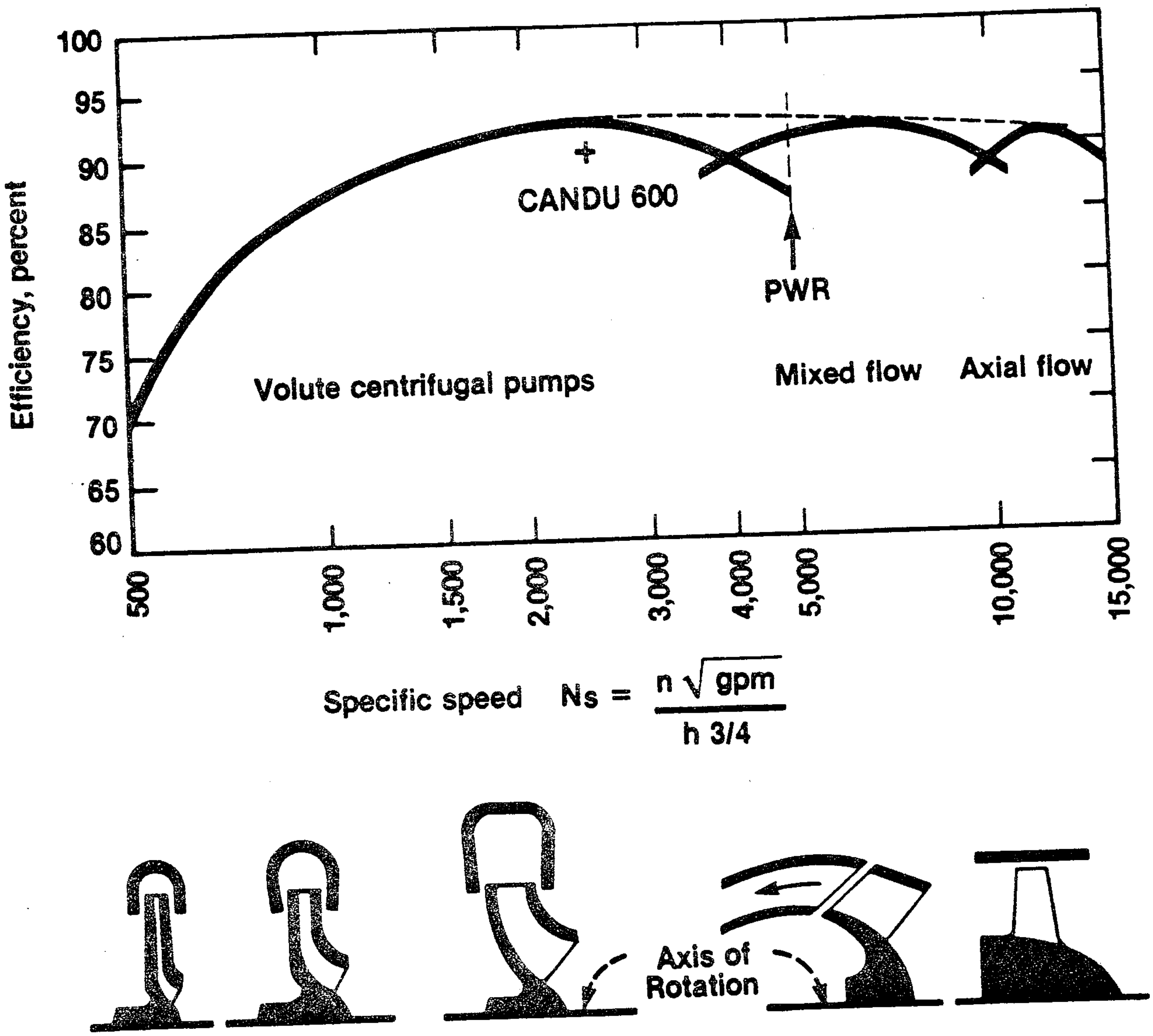


FIGURE 17 HEAT TRANSPORT PUMP SPECIFIC SPEED

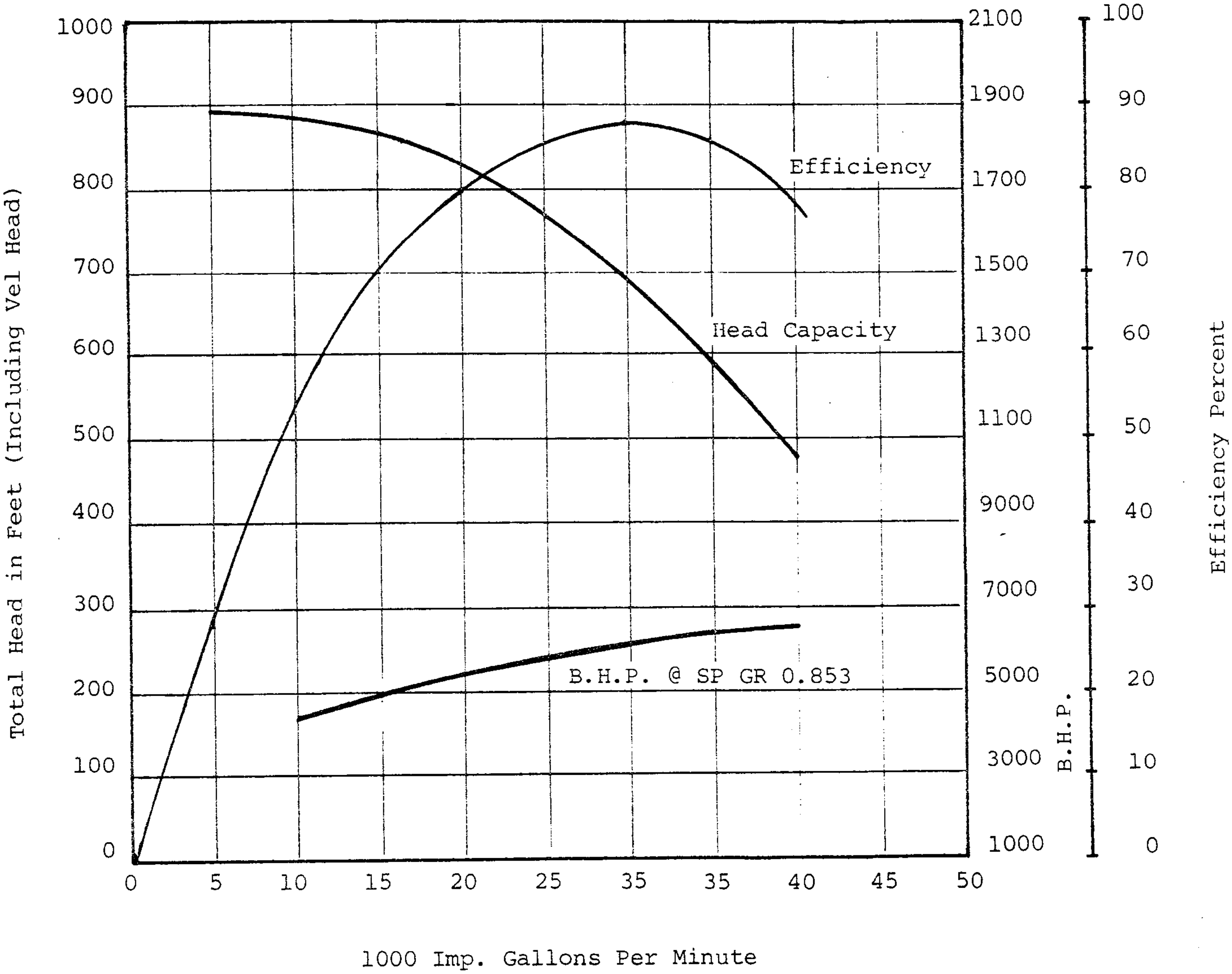


FIGURE 18

PRIMARY PUMP CHARACTERISTIC CURVE

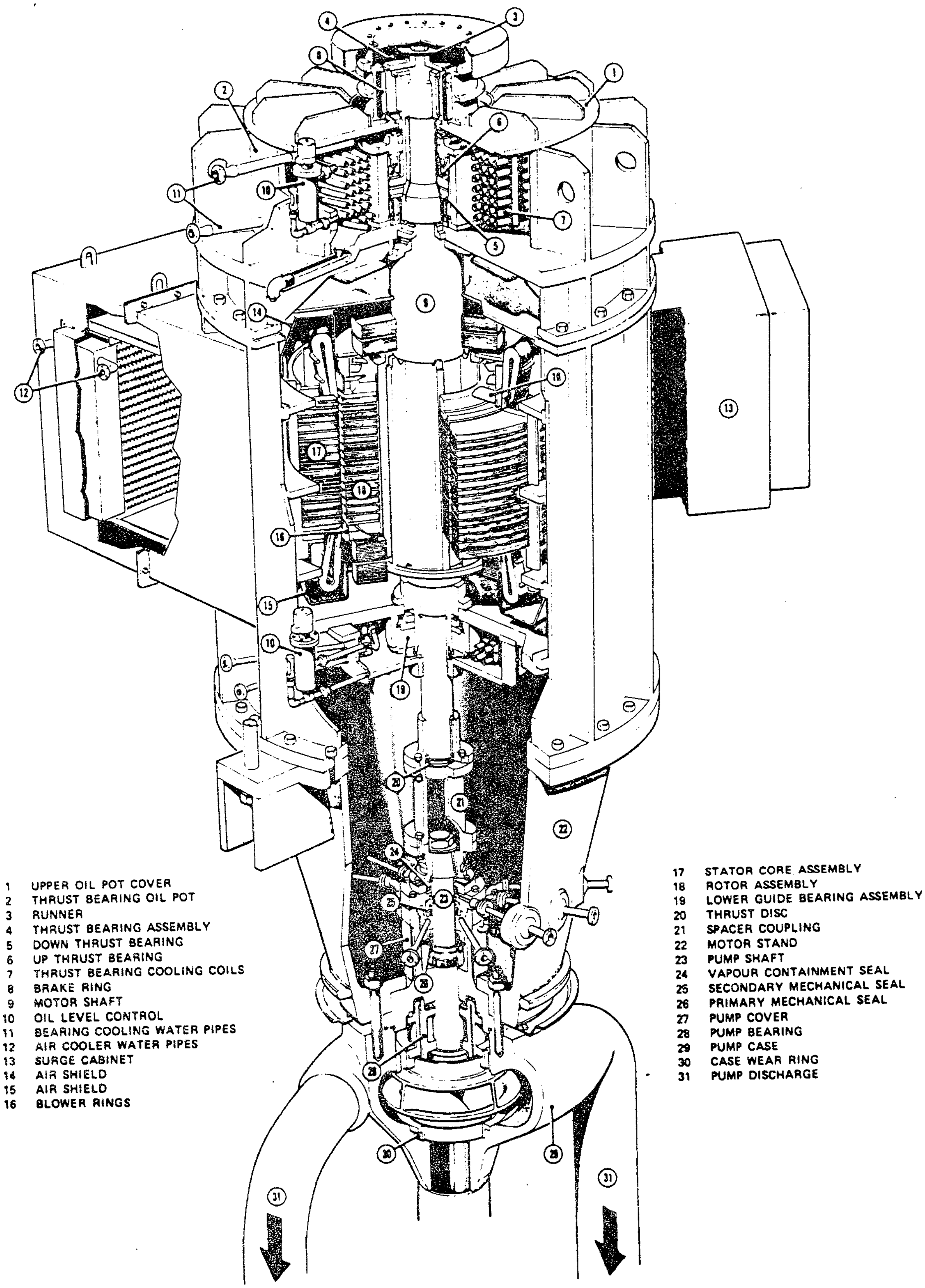


FIGURE 19 TYPICAL CANDU 600 HEAT TRANSPORT PUMP

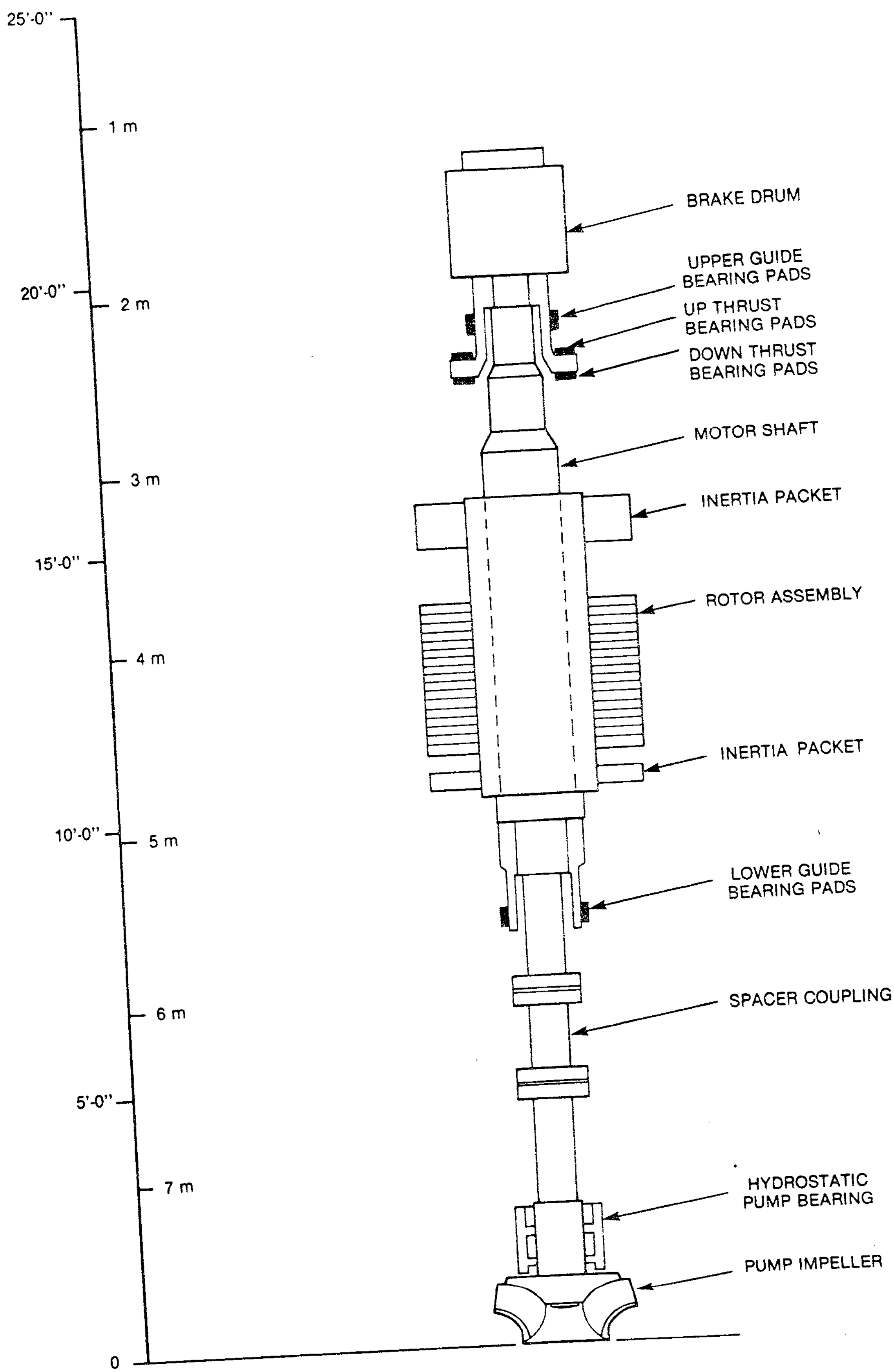


FIGURE 20 TYPICAL HEAT TRANSPORT PUMP MOTOR ROTOR SYSTEM

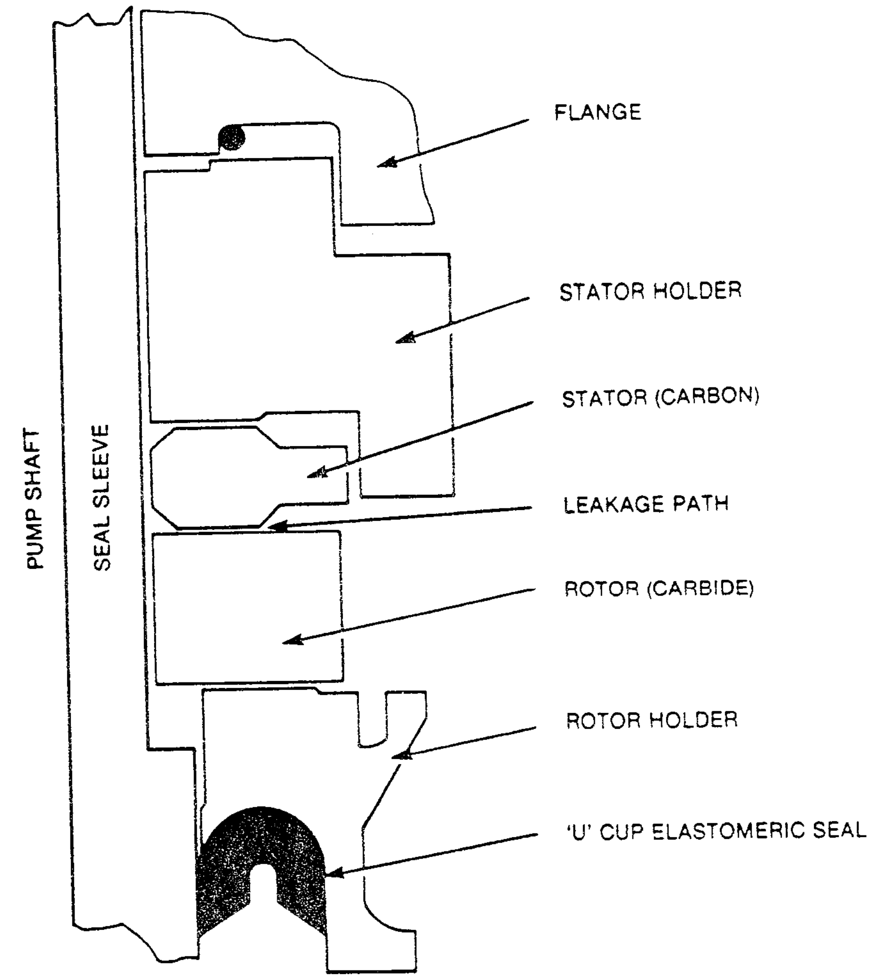


FIGURE 21 SEAL OPERATING FACES

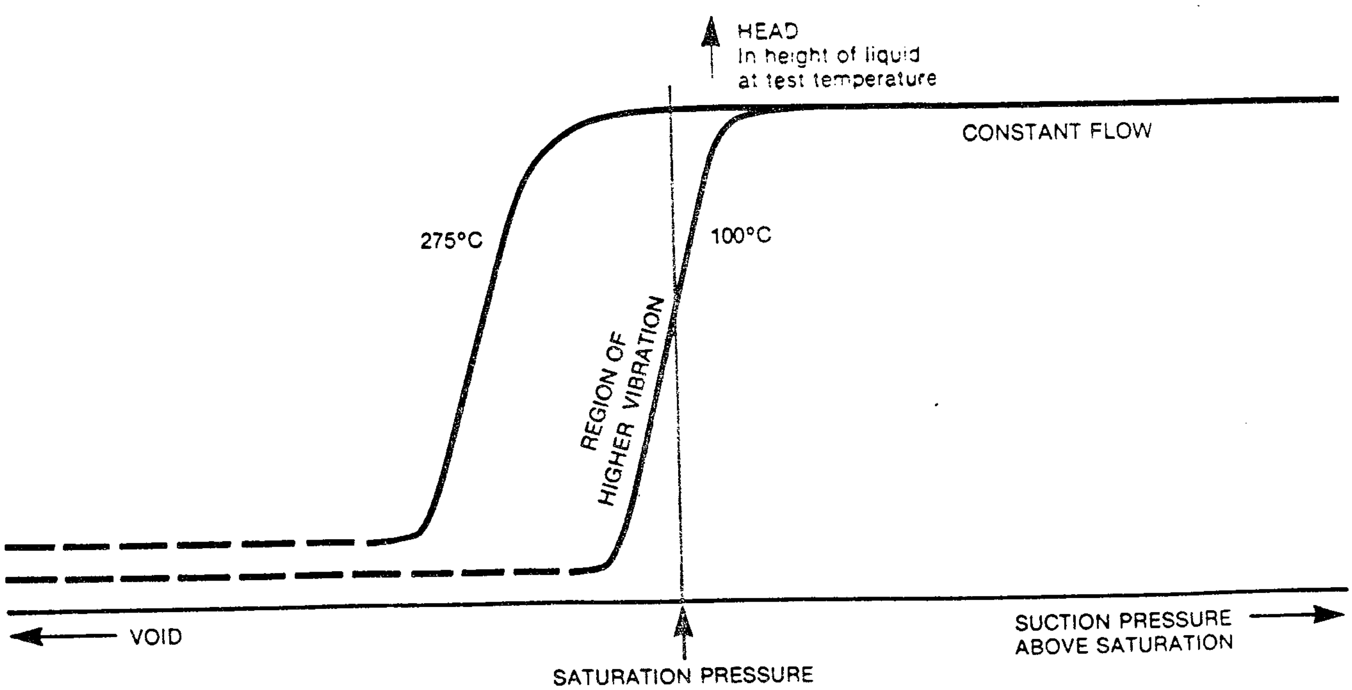


FIGURE 22 PUMP TESTING AT LOW SUCTION PRESSURES (DIAGRAMMATIC)