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**High Pressure Pumps, Drivers, and Energy
Recovery Devices for Large Seawater
Reverse Osmosis Plants**

OAK RIDGE NATIONAL LABORATORY
OPERATED BY UNION CARBIDE CORPORATION · FOR THE DEPARTMENT OF ENERGY

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HIGH PRESSURE PUMPS, DRIVERS, AND
ENERGY RECOVERY DEVICES FOR LARGE
SEAWATER REVERSE OSMOSIS PLANTS

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ABSTRACT

The primary objective of this study was to prepare a technical survey of currently available high pressure pumps, drivers, and energy recovery devices for large reverse osmosis (RO) plants. This equipment was specified for two seawater RO plant sizes (2.5 MGD and 25 MGD) and two product recovery ratios (25 and 50 percent) operating at 900 psi pressure. Four types of drivers were considered; electric motors, steam turbines, gas turbines, and diesels. It was concluded that: pumps suitable for seawater RO systems are available in single units for plant sizes through 25 MGD; the cost and availability of energy is the most significant item in the selection of a pump driver; energy recovery can be an economic choice with a low recovery ratio and high values of energy cost, operating pressure, and plant size. Also, since the cost of power can be a large part of the annual cost, an optimization of the pumping system is worthwhile.

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Section 1

INTRODUCTION

Reverse osmosis (RO) is a process for desalting saline water by supplying hydrostatic pressure to drive water to be purified through a semipermeable membrane. Most of the dissolved ions in the water remain behind and are discharged in the brine reject stream. Although the application of reverse osmosis technology to seawater desalting is a recent development, it promises to become a major contributor to the amount of potable water produced from the sea. It offers the advantages of more efficient energy utilization and potentially lower capital cost than existing seawater thermal desalting techniques. Seawater RO plants as large as 3.2 MGD are in operation.

A membrane system comprises the following process steps: feed pretreatment, pressurization of the feed stream, the RO modules, energy recovery from the brine stream, and post treatment of the product. The major power required is for the high-pressure feed pumps. The feed pumping and energy recovery system represent a major factor in the system design as:

- 1) The high pressure pump represents the largest use of power in the system
- 2) The power consumption can be the largest single operating cost
- 3) A high on-stream factor can be obtained by proper system design

- 4) This system is one of the largest capital cost items in the total plant.

The importance of the pumping system was recognized early in the development of the reverse osmosis process as the Office of Saline Water (OSW) sponsored a study, which was published in 1969, on "Pumping and Energy Recovery Ssystems for Reverse Osmosis (RO) Desalination Plants" (Ref. 1-1). This study provided information on principal operating characteristics of the equipment, on the economic and technical feasibility of energy recovery, and on the areas of development required for both pumping and energy recovery systems. This 1969 report has since been outdated with changes in:

- RO system operating characteristics
- pumping equipment available
- costs, especially energy.

Bechtel undertook a study of a 25 million gallon per day (MGD) combined nuclear-electric seawater RO desalting plant for Union Carbide Corporation in August 1977. This work was accomplished for Union Carbide Corporation under Contract Number ORNL 11-Y-13511-V and was reported in Reference 1-2. One of the recommendations from that study was that a more detailed review be made of the current status of high-pressure pumps as well as energy recovery devices.

1.1 OBJECTIVE

The primary objective of this report was to prepare a technical survey of currently available high pressure pumps, drivers, and energy recovery devices for large RO desalination plants. This is to update the information in the referenced OSW report. In addition, cost information on these devices was to be obtained wherever possible.

1.2 APPROACH

The approach followed was to establish study ground rules based upon recent design requirements for seawater reverse osmosis systems. Two large RO plant sizes were selected, each operating at two different recovery ratios. For these four cases, a single pump and driver were specified. Manufacturers were then contacted to provide information on cost, availability, and performance characteristics. This was also done for energy recovery turbines in a similar manner. Using this information, the cost of product water as influenced by the pump and energy recovery system was examined.

Section 2

SUMMARY AND CONCLUSIONS

2.1 SUMMARY

Pumps, drivers, and energy recovery turbines were selected for two seawater reverse osmosis plant sizes (2.5 MGD and 25 MGD) and two product recovery ratios (25 and 50 percent) operating at 900 psi pressure. Single items of equipment were specified for each of these pieces of equipment, and manufacturers were contacted to obtain price, availability, and performance information.

2.1.1 Pumps

Pump specifications were developed based upon the given plant sizes, recovery ratios, pump discharge pressure of 900 psi, and 316 SS as the material of construction. Six pump manufacturers were contacted and supplied information which is summarized in Table 2-1.

Pumps as single units are available for all four selected applications. Efficiencies near the maximum practical attainable value were quoted by the manufacturers. Pump delivery times of up to two years can be expected for the largest size pumps. Pump speeds varied between 1800 and 7600 rpm. The high speed pumps

DESCRIPTION	2.5 MGD PLANT	25 MGD PLANT
Efficiency (%)	81 - 87	83 - 89
Pump Flow (gpm)	3470 - 6940	34,700 - 69,400
Pump Speed (rpm)	3560 - 7650	1780 - 4450
Unit Price (\$/gpm of feed)	24 - 25	15
Delivery Time (Weeks)	32 - 104	40 - 104
Reference Section	4.2	4.2

TABLE 2-1. PLANT PUMP SUMMARY - 2.5 and 25 MGD

offer advantages due to the smaller size, fewer number of stages, and better speed matching characteristics with some drivers.

2.1.2 Drivers

Specifications were developed for four types of drivers --- electric motors, steam turbines, gas turbines, and diesels. The information is summarized in Table 2-2.

Information was obtained on electric motors for three of the four driver applications. The largest size motor (about 43,000 horsepower) requires either a custom designed induction motor, two standard size motors in series, or use of a synchronous motor. Engine efficiency is very high (about 96 percent) for motors of this size.

Steam turbines offer the advantage of high horsepower, variable speed, and high efficiency and reliability. Steam turbines are available in sizes for all the pump applications.

Gas turbines offer advantages as pump drivers due to their high operating speed and ability to use a variety of fossil fuels, especially natural gas. No information was received for the smallest size turbine (about 2200 Hp). The largest size turbine for which information was received may require the use of a helper turbine to provide the required pump horsepower.

DESCRIPTION	PLANT SIZE (MGD)	DRIVER			
		ELECTRIC MOTOR	DIESEL	GAS TURBINE	STEAM TURBINE
Efficiency (%)	2.5 ↓	95 - 96	35 - 37	21 - 27	56 - 69
Efficiency Type		Engine	Thermal	Thermal	Engine
Unit Price (\$/Hp)		19 - 25	160 - 180	260	130 - 140
Delivery (Weeks)		—	29 - 104	43 - 78	52 - 75
Efficiency (%)	25 ↓	95 - 95	38	23 - 28	71 - 74
Unit Price (\$/Hp)		13	290 - 300	75 - 130	26 - 43
Delivery (Weeks)		—	104	60 - 70	60 - 75
Reference Section	2.5 & 25	5.1	5.4	5.3	5.2
Variable Speed		No	Yes	Yes	Yes
Source of Energy		Electricity	Various Fossil Fuels	Various Fossil Fuels	Steam

TABLE 2-2. PUMP DRIVER SUMMARY - 2.5 and 25 MGD

Diesel engines are one of the most efficient internal combustion devices. They have high reliability and can use a variety of fossil fuels. These devices are characterized by their slow operating speed. Information was received for three of the four applications; the largest size would require two engines in tandem service.

Some of the most important factors to be considered in the selection of a driver include the capital cost, efficiency, and fuel cost. The capital costs shown in Table 2-2 are for the driver only and not the support equipment. In some cases, this support equipment can add considerably to the driver cost. Two types of driver efficiency are shown — engine efficiency and thermal efficiency. These cannot be directly compared due to different bases of evaluation. Energy costs vary widely depending upon fuel availability and local price regulations.

2.1.3 Energy Recovery

Since the fresh water recovered through the membrane is only a fraction of the feed flow of seawater, there is a large high pressure brine stream available for energy recovery.

Specifications were developed for impulse turbines for the energy recovery device. Impulse turbines offer high efficiency. In smaller size applications, the price is about the same regardless of flow. This information is summarized in Table 2-3.

DESCRIPTION	2.5 MGD PLANT	25 MGD PLANT
Unit Price (\$/Hp)	115 - 320	47 - 85
Efficiency (%)	88 - 90	89 - 90
Delivery Time (Weeks)	80	130
Reference Section	6.2	6.2

TABLE 2-3. ENERGY RECOVERY TURBINE SUMMARY -
2.5 and 25 MGD PLANT

2.1.4 System Selection

A complete RO system includes a pump and driver and, possibly, the use of an energy recovery system. Selection of the pump and driver are related to factors such as reliability, compatibility of the two devices, load schedule of the plant, but most importantly, to the cost and availability of energy for the driver.

Their selection, therefore, depends upon site specific information. Energy recovery is also related to the cost of energy, as well as the plant size, operating pressure, and recovery ratio. Two energy recovery cases are discussed.

For a particular case, studied in Section 7 for a 2.5 MGD plant, the "break-even" point varies between 0.023 and 0.033 \$/Kw-Hr for 25 and 50 percent recovery ratio plants. At power costs below these values, energy recovery is not worthwhile, while above this rate, energy recovery should be considered. Larger size plants can be expected to show "break-even" points at even lower power costs due to the lower unit costs for larger size energy recovery turbines.

2.2 CONCLUSIONS

1. Pumps suitable for seawater RO systems are available in single units for plant sizes through 25 MGD. Single units may not be desirable due to the effect on reliability and control so that a multiple pump installation may be more desirable.
2. The efficiency of pumps increases with flow. Pumps are available for seawater that approach the maximum practical efficiency attainable.
3. The cost and availability of energy is the most significant item in the selection of a pump driver. Generally, the annual energy cost for operation is much more than the annualized equipment cost.
4. Energy recovery can be an economic choice with high values of energy cost, operating pressure, and plant size, and a low value of energy recovery ratio (e.g. 25 percent as compared to 50 percent).
5. An engineering study must be made for an appropriate pump, driver, and energy recovery system for each specific site. Since the cost of power can be a large part of the annual cost, an optimization of this system is worthwhile.
6. The delivery times for equipment in the pump system can be a significant factor in determining the construction schedule for the plant.

Section 3

BACKGROUND

3.1 PREVIOUS WORK

A comprehensive review of pump and energy recovery systems was made for the Office of Saline Water in 1969 (Ref. 1-1). The purpose of the study was to determine the state-of-the-art with respect to pumping equipment, and the technical and economic feasibility of recovering energy from RO systems.

The study concluded that the cost of energy (steam or electricity) should be carefully evaluated at each plant site to select the most economical driver. On the basis of energy costs used in the study, steam-turbine drivers produced the lowest pumping system costs for the larger desalination plants (1 MGD to 10 MGD). The driver selection was less critical for 0.1 MGD plants. For large plants, multiple pumps were used, since no single large pump was available for 10 MGD applications. The report recommended further development of large seawater pumps by the pump manufacturers.

The study also concluded that cost savings resulting from adding an energy recovery device to the system depended on the following parameters, in order of descending importance:

- fresh water production rate
- fresh water recovery factor (product flow/feed flow)

- membrane pressure level
- efficiency of the energy recovery system and type of energy recovery arrangement
- service and maintenance costs
- good matching of the various shaft speeds.

A comparison of the ground rules used in that study with more typical present operating practice, is shown in Table 3-1.

The changes in the system operating conditions have come about due to major advances in membranes. In addition, the cost of energy has increased substantially (over four times) since 1969. Comparing plant construction costs, shown in Table 3-1, it is seen that they have doubled since 1969.

3.2 STUDY GROUND RULES

Shown in Figure 3-1 are the main components of the pump and energy recovery system. These are the pump driver, the pump, and the energy recovery device. The amount of power required is determined by the product flow, recovery ratio, operating pressure, and the efficiency of the various devices. Two product flows were selected that are representative of the range of plant sizes expected in the near future; they are 2.5 and 25 MGD. Also, two recovery ratios, 25 and 50 percent, were selected which represent the typical range of operation of seawater RO desalting plants. (It should be noted that a

PARAMETER	1969 Previous Study (Ref. 1-1)	1978 Typical Conditions
Operating Pressures ~ psi	1500	800 - 1000
Recovery Ratio ~ %	40	25 - 50
Membrane Life - years	0.5	2+
Plant Size ~ MGD	0.5, 1, 10	3
Energy Cost: $\$/10^6$ BTU	0.55	2.50
$\$/Kw-Hr$	0.007	0.025
Drivers Considered	Electric motor Steam turbine Diesel engine	Same + Gas Turbine
Construction Costs:		
Chemical Engineering Plant Cost Index (Ref. 3-1)	119	220
Marshall and Swift Cost Index, Process Industry	285	560

TABLE 3-1. COMPARISON OF ECONOMIC
AND TECHNICAL REQUIREMENTS
FOR SEAWATER RO SYSTEMS

Figure 3-1
REVERSE OSMOSIS PUMP AND ENERGY RECOVERY SYSTEM

50 percent recovery ratio requires half the feed flow of a system operating at 25 percent recovery ratio for the same product flow.)

A single pump was specified for each plant. Thus, four different cases are considered and are shown in Table 3-2. It should be noted that in the event multiple pumps are used, this information can also be applied. For example, two pumps of Case 4 or ten pumps of Case 1 can be used to satisfy the requirements of Case 3.

Information was also to be obtained on four types of pump drivers. Three of these were considered in the referenced report — electric motor, steam turbine, and diesel. A gas turbine driver was added for this study.

The ground rules and operating conditions for this study are summarized in Table 3-2.

TABLE 3-2 STUDY GROUND RULES

- Plant sizes - 2.5 and 25 MGD product water capacity
- Pump discharge pressure - 900 psig
- Recovery ratios - 25 and 50 percent
- Pump type - centrifugal
- Drivers considered -
 - electric motor
 - steam turbine (with steam conditions typical of a nuclear power plant)
 - gas turbine
 - diesel
- Energy recovery units
 - turbine type
 - others
- Driver coupling
 - electric
 - hydraulic

CASE NO.	1	2	3	4
Plant Size MGD	2.5		25	
Recovery Ratio %	25	50	25	50
Pump Flow gpm	6940	3470	69,400	34,700
Brine Flow gpm	5210	1730	52,100	17,300

Section 4

CENTRIFUGAL PUMPS

4.1 INTRODUCTION

Centrifugal pumps are a type of pump in which energy is imparted to the pumped liquid by means of a moving impeller or propeller rotating on a shaft. The velocity energy imparted to the fluid by the moving impeller is later partially converted to pressure energy. These pumps are divided into two classes, horizontal and vertical, and can be subclassified as radial flow, mixed flow, axial flow, and regenerative turbine. They are characterized by the fact that flow can be varied from near zero to a maximum design value with a resultant change in the developed head and power requirement.

The other major type of pump is positive displacement. They are not considered in this report since they are not generally used in the flow and pump head range specified.

4.2 OPERATION

Pump horsepower can be calculated by the following relationship:

$$\text{BHP} = \frac{Q \times H \times \text{SG}}{(3960) \times \eta_p}$$

where Q(GPM) is the flow of the pumped liquid, H(ft) is the differential head across the pump, SG is the specific gravity

of the pumped liquid, and η_p is the efficiency of the pump. The four design flow rates specified in the ground rules and the approximate horsepower required is shown in Table 4-1:

TABLE 4-1. PUMP SIZES

<u>Case No.</u>	<u>RO Plant Size</u>	<u>Pump Designation</u>	<u>Recovery Ratio %</u>	<u>Flow (GPM)</u>	<u>Approximate Pump Horsepower (HP)*</u>
1	2.5 MGD	P-1	25	6940	4300
2	2.5 MGD	P-2	50	3470	2200
3	25 MGD	P-3	25	69400	43000
4	25 MGD	P-4	50	34700	21500

*Assuming 80 percent pump efficiency and 900 psi differential pressure (2025 feet of head).

Centrifugal pumps are available in a wide variety of flow and head capabilities with the largest centrifugal pump having flow capacities of 90,000 GPM at 3000 ft head. To obtain optimum performance, a multi-stage pump, or the use of pumps operating in parallel or series, is often required.

4.3 PERFORMANCE

Pumps, in general, can be classified by the quantity specific speed, N_s , defined as:

$$N_s = \frac{N \times \sqrt{Q}}{H^{3/4}}$$

where:

N = rotation speed (rpm)

Q = flow (gpm)

H = differential head/stage (feet)

(All values at pump best efficiency point.)

The efficiency as it relates to specific-speed for a centrifugal pump is shown in Figure 4-1. This chart is useful in the selection of pumps with maximum efficiency.

To obtain a pump for a given flow and head with maximum efficiency, either the number of stages or pump speed can be changed to obtain an optimum value of specific speed. In boiler feed pump applications (characterized by high head operation), the trend in design has been toward higher pump speeds than those obtained with electric motor drives (Ref. 4-1). Pump speeds of 5000 to 6000 rpm are common, which provides the advantage of a reduction in the number of pump stages and in pump size. This reduces the pump shaft span and deflection. This also allows direct coupling of the pump to steam turbines which typically operate at these speeds.

Figure 4-1 Single Stage Centrifugal Pump Efficiencies (Ref. 1-1)

Maximum system efficiency can be obtained by matching the pump head and flow to the desired operating conditions. Centrifugal pumps obey the affinity law:

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \sqrt{\frac{H_1}{H_2}}$$

The differential head varies with the pump speed; thus, the same pump can be used to obtain higher heads and flows by increasing the speed. Therefore, the pump output can be matched to the system operating conditions if a variable speed drive is available. If constant speed operation is used, the throttling of the pump discharge is required, which results in a waste of pump power.

All pumps require some net positive suction head (NPSH) to prevent cavitation which reduces efficiency and causes damage to the pump. Cavitation results in the formation of vapor bubbles in the liquid, which then quickly collapse, releasing energy that attacks the impeller vane or shroud. The NPSH value of the pump is influenced by the speed of rotation and the inlet area of the impeller. The NPSH required is often provided by placing the pump below the liquid level or by using a booster pump.

Because of the corrosive nature of seawater, appropriate materials of construction must be selected. Materials such as 316 stainless

steel, Ni Resist, or bronze have been used for seawater pumping system. Stainless steel was specified in the pump specifications.

4.4 SPECIFICATIONS

Based upon the information shown in Table 4-1, a pump specification was developed. A standard pump specification was utilized, specifying a pumped fluid temperature of 80°F, a discharge pressure of 900 psi (2025 ft. based upon 0 psi inlet pressure), and use of 316 SS as the preferred material of construction. This was sent to six pump manufacturers with a request for price, delivery, and performance information.

4.5 RESULTS

Pumps for Cases 1, 2, and 4 (pumps operating up to 35,000 GPM @ 2025 ft head) are readily available from pump manufacturers.

The pump for Case 3 in this study (P-3, operating at 69,400 GPM @ 2025 ft head) is available from a limited number of pump manufacturers. Sulzer Bros.* and Ingersoll-Rand* were contacted and provided information for this large pump; although some of the other manufacturers indicated they could provide estimates for a specific application. In all cases, a single-stage or multi-stage pump was provided by at least one manufacturer (as opposed to parallel/series pump operation).

* Suppliers names and addresses are in the Appendix

A net positive suction head of 50 - 300 ft was necessary for most of the pumps suitable for P-1 through P-4. Booster pumps are required to obtain this NPSH. The booster pump will add to the initial pumping costs, particularly those requiring both a high flow and a relatively large differential head.

One of the major operating characteristics of the pumps is the pump speed. A great variation of speeds, from 1800 rpm to 7500 rpm, are found in various pumps meeting the specification requirements. This speed variation is important as some pumps may be more compatible with some types of drivers without the need for extensive gearing. In addition, the speed affects cost and reliability.

A summary of the performance characteristics is shown in Tables 4-2 through 4-5. A comparison of the actual pump efficiency with that shown in Figure 4-1 shows that the actual efficiency is very near the maximum value to be expected.

Manufacturer	UNITED	UNITED	INGERSOLL-RAND	WORTHINGTON	BINGHAM	SULZER
Model/Size	10 x 18 BFD	J-10 x 17 DVS	10 x 23 PH	14 WGPL-121	12 x 12 x 17A	HP 40 e + e
# Stages	2	1	2	1	2	2
Speed	3560	5000	3570	7650	3560	3600
Efficiency %	80	83	83	87	86	85
NPSH	55	98	40	150	60	105
Delivery Time - wks.	36	32	52 - 104	—	—	70

2.5 MGD at 25% Recovery (2025 ft at 6940 GPM)

TABLE 4-2. DESCRIPTIVE RESULTS - PUMP P-1

Manufacturer	UNITED	UNITED	INGERSOLL-RAND	WORTHINGTON	BINGHAM
Model/Size	T 8 x 13 WMSND	U 6 x 11 WMSNDH	8 x 13 DA-3	10 WGPL 111	10 x 10 x 13½ B
# Stages	3	3	3	1	3
Speed	3560	4500	3560	7600	3560
Efficiency %	81	81	84	82	84
NPSH	60	80	30	75	50
Delivery Time - wks.	34	34	52 - 104	—	—

2.5 MGD at 50% Recovery (2025 ft at 3470 GPM)

TABLE 4-3. DESCRIPTIVE RESULTS - PUMP P-2

Manufacturer	INGERSOLL-RAND	SULZER
Model/Size	30 x 40 JM	HP 85 ede
# Stages	2 pumps in series	2
Speed	1780	1800
Efficiency %	84	88
NPSH	150	157
Delivery Time - wks.	52 - 104	70

25 MGD at 25% Recovery (2025 ft at 69,400 GPM)

TABLE 4-4. DESCRIPTIVE RESULTS - PUMP P-3

Manufacturer	UNITED	INGERSOLL-RAND	WORTHINGTON	BINGHAM	SULZER
Model/Size	24 x 34 BFD	20 x 36 JM	30 WGPL 211	24 x 28 x 28	HP 80 ede
# Stages	2	2 pumps in series	1	—	2
Speed	1780	1780	4450	3560	1800
Efficiency %	83	85	86	89	87
NPSH	102	90	270	300	72
Delivery Time - wks.	40	52 - 104	—	—	70

25 MGD at 50% Recovery (2025 ft at 34,700 GPM)

TABLE 4-5, DESCRIPTIVE RESULTS - PUMP P-4

Section 5

DRIVERS

Four types of pump drivers were considered. They are electric motor; steam turbine; gas turbine; and diesel engine.

Four cases of driver size were specified based upon the information shown in Table 4-1. The driver sizes are predicated upon an assumed pump differential head of 900 psi and efficiency of 80 percent. In a final determination of the driver size, the actual pump differential head and pump and coupling efficiency are required.

One of the most important factors in driver selection is the energy supply due to large variability in cost and availability. The final selection of a pump driver will also depend upon cost, reliability, delivery time, and the driver operating characteristics as related to the pump requirements.

5.1 ELECTRIC MOTORS

5.1.1 Introduction

Alternating current electric motors are commonly used for pump drivers. These are available as either induction or synchronous motors. Electric motors are one of the simplest and most reliable of all drivers. These motors consist of two basic electric assemblies; the wound stator and the rotor assembly.

Electric motors are available in standard sizes; up to 20,000 horsepower for induction motors, and up to 50,000 horsepower for synchronous motors. Custom designed units are available in larger sizes.

5.1.2 Operation

Alternating current motors are inherently constant speed devices. The synchronous speed of the motor (the speed which the magnetic field rotates) is determined by the number of poles in the stator and the frequency of the power supply as follows:

$$N_s = 120 \frac{f}{p}$$

where N_s (rpm) is the synchronous speed, f (Hz) is the frequency of the alternating current, and p is the number of poles in the motor. The synchronous speed is attained by synchronous motors, but with induction motors the rotor rotates slower than the magnetic field. This slippage is required as there is no torque developed by the motor when the rotor and magnetic field rotate at the same speed. Thus, a two-pole synchronous motor operating on 60 Hz current operates at 3600 rpm, while an induction motor might operate at 3575 rpm.

5.1.3 Performance

Electric motor efficiency is primarily a function of size. Small universal motors (in the range of fractions of a horsepower) have efficiencies of only 60 percent. However, large electric motors

of the size required for the plants specified, have very high efficiencies in the range of 96 percent. The unity power factor synchronous motor generally has higher efficiency than the squirrel cage induction motor. Efficiency varies with load and is highest at the rated load.

In addition to efficiency, electric motors are rated by their power factor. The dimensionless quantity is the ratio of active power to apparent power. For large squirrel cage induction motors, the power factor is usually about 90 percent at full load. Thus, the system must be designed for larger current requirements than are actually used. This may result in an increased charge for electricity. Synchronous motors can be obtained with either 100 percent or 80 percent power factors.

Factors to be considered in the installation of electric motors are the supply voltage available, starting current requirements, and motor protection enclosure required, as they influence the cost of the installation. Large starting currents, i.e. six times full load current, may be required. Various types of motor protection are available depending upon the motor's environment. Among the enclosure types are drip-proof, weather protected, and totally enclosed.

5.1.4 Specification

A horizontal, squirrel cage induction motor was specified for the pump driver. A weather protected type II enclosure was specified with a motor speed of 3600 rpm (if available), and the manufacturer's standard voltage for the required motor size.

5.1.5 Results

Two electric motor manufacturers were contacted to provide information — General Electric and Westinghouse. AC electric motors (squirrel cage induction type) are available to meet the pump driver requirements in Cases 1 through 4. For Case #3 (43,000 BHP driver), a configuration of two motors operating in tandem is required instead of the usual single motor operation. Tandem operation is not an unusual practice, but uneven load sharing between the motors may cause problems. Tandem operation is required because induction motors are not normally available above 20,000 HP as standard size units, although synchronous motors are available in excess of this size.

A summary of the electric motor results is given in Table 5-1. Motor speed has a significant effect on price. The maximum available speed of 3600 rpm was selected except for the largest size motors where a maximum of 1800 rpm was available. The efficiency of the motors in the size range considered was 96 percent (synchronous motors have slightly high efficiencies).

Case No.	Motor Size Horsepower	Voltage	Speed RPM	Efficiency %
1	4,500	4000	3600	96
2	2,300	2300	3600	96
3	43,000	not available	not available	not available
4	20,000	4000	1800	96

TABLE 5-1. RESULTS - ELECTRIC MOTORS

5.2 STEAM TURBINES

5.2.1 Introduction

Steam turbines are often installed in applications requiring large horsepower, variable speed, and high efficiency and reliability. Steam turbines can be designed to operate under almost any steam conditions. Because of this flexibility, it is common for the availability of steam from within a plant to influence the selection of a steam turbine driver. Compared to other prime movers, steam turbines occupy less floor space, utilize lighter foundations, require no internal lubrication system, and require less attendance by an operator. Steam turbines offer high reliability and low maintenance costs.

5.2.2 Types of Steam Turbines

Steam turbines range from a few horsepower to as much as one million horsepower. Various types of steam turbines are available, operating in condensing or non-condensing modes. In condensing turbines, steam leaves the turbine exhaust at a pressure below atmosphere, while in non-condensing turbines, steam leaves the turbine exhaust at atmospheric pressure or above.

A non-condensing steam turbine is used when relatively high temperature and pressure steam can be purchased and returned, sold, or used within a plant at a reduced temperature and pressure. This type of steam turbine combines high operating costs

and relatively high consumption of steam per horsepower delivered, as the steam is not reduced to a very low pressure. Because no steam is condensed, the capital cost of a condenser is not required and the expense of maintaining a supply of cooling water is eliminated; however, this is at the expense of increased feedwater treatment costs. The steam turbine is smaller in size and less expensive because the exhaust is not at a very low pressure.

A condensing steam turbine can use inlet steam having the same temperature and pressure as a non-condensing unit. However, a condenser is utilized to condense the exhaust steam, which can then be pumped back at high pressure to the boiler to be reheated (closed system).

A well-designed condensing steam turbine, utilizing a greater proportion of energy in the steam, should be able to operate with as little as 50 percent of the steam required by a non-condensing turbine having the same output. This is offset by its higher initial cost, the cost of the condenser, and the cost of maintaining a cooling water supply.

5.2.3 Performance

The efficiency of steam turbines can be expressed as:

$$\text{engine efficiency} = \frac{\text{actual adiabatic work}}{\text{reversible adiabatic work}}.$$

A general estimate of efficiencies to be expected is shown in Figure 5-1 for various turbine sizes and operating conditions. It is apparent turbine efficiency increases with size. Thus, in cases of smaller pump power requirements, the overall efficiency of electric power generation (combined turbine, generator, and electric motor) can be greater than a direct steam turbine drive.

Efficiency can also be expressed in terms of the "steam rate". The theoretical steam rate can be calculated from enthalpy data for an ideal expansion process. The corresponding actual steam rate can be obtained by dividing the theoretical steam rate by the (engine) efficiency to account for thermodynamic and mechanical losses.

5.2.4 Specifications

In purchasing a steam turbine, one must specify the inlet steam conditions (temperature and pressure), the discharge pressure, the power to be delivered, and the preferred shaft speed. In this study, the inlet condition was saturated steam at 1015 psig @ 548°F while the exhaust pressure was 3.5 inches of mercury (absolute). These conditions are typical for a nuclear power plant (pressurized water reactor) as outlined in the ground rules.

The turbine efficiency and investment costs increase with high inlet steam pressures and temperatures and lower exhaust pressures.

FIGURE 5-1. STEAM TURBINE EFFICIENCIES
vs. POWER FOR VARIOUS INLET
STEAM CONDITIONS (Ref. 1-1)

5.2.5 Results

Steam turbines, manufactured in a wide range of sizes, are available to meet the pump driver requirement of the four cases studied in this report. The speeds of the turbines vary from 3600 to 6000 rpm. The efficiencies of the turbines varied between 56 percent and 74 percent. The small turbines were usually less efficient. The steam rate of the turbines varies from 8.7 lb/BHP-hr to 11.6 lb/BHP-hr. A summary of the steam turbine description information is shown in Table 5-2.

5.3 GAS TURBINES

5.3.1 Introduction

Gas turbines offer many advantages as pump drivers. Some of these advantages are:

- low cost per horsepower
- compactness
- use of a variety of fuels
- high operating speed.

Disadvantages include poor fuel efficiency and lower reliability than steam turbines.

5.3.2 Operation

The driving fluid in a combustion gas turbine is a heated gas. The gas turbine unit usually consists of compressors, combustors,

DRIVER	DP-1	DP-1	DP-1	DP-2	DP-3	DP-3	DP-4	DP-4	DP-4
CASE NO.	1			2	3		4		
REQUIRED POWER	4300 Hp			2250 Hp	43,000 Hp		32,500 Hp		
MANUFACTURER	ELLIOT	G.E.	G.E.	ELLIOT	ELLIOT	G.E.	ELLIOT	G.E.	G.E.
MODEL	ZEPG 7			ZEPG 7	2NV110F		2NV9		
POWER OUTPUT Hp	4,300	5,000	5,000	2,200	43,000	43,000	21,500	21,500	21,500
SPEED rpm	3,600	3,600	6,000	3,600	3,600	6,000	3,600	3,600	6,000
EFFICIENCY %	58.4	68.7	68.2	55.6	72.0	71.2	71.4	73.8	72.8
STEAM RATE lb/Hp-Hr	11.0	9.4	9.5	11.6	8.95	9.1	9.0	8.7	8.9
DELIVERY PERIOD WEEKS	52	75	75	52	70	75	60	75	75
SIZE (Inches) L x W x H	100 x 86 x 70	135 x 148 x 108	105 x 96 x 96	100 x 86 x 70	180 x 130 x 95	178 x 110 x 137	165 x 130 x 95	161 x 148 x 143	162 x 110 x 117

TABLE 5-2. RESULTS - STEAM TURBINES

and turbines. Compressed air is delivered to a combustion chamber from a rotary compressor driven by a direct connected turbine. The exhaust and heated air from the combustor expand through the turbine, delivering power in excess of that required for the air compressor. For a simple-cycle gas turbine, approximately two-thirds of the work available from the expanding exhaust gases is used to run the air compressor, leaving one-third of this work for the turbine output. The exhaust gases are at a high temperature so that they are available for regeneration or for steam production in a combined cycle plant for increased cycle efficiency.

5.3.3 Types of Gas Turbines

Gas turbines are manufactured in a relatively wide power and speed range. They are available in a power range from approximately 100 HP to about 50,000 HP with shaft speeds up to 10,000 rpm. There are two types of gas turbines:

- Open systems
 - working fluid (air and combustion products) pass through the plant only once
 - advantages of this mode are simple requirements for control and no cooling water requirements
- Closed systems
 - working fluid is continuously recycled
 - heat from external source is transferred through the walls of a closed heater

- advantages of this mode of operation include clean working fluid, constant pressure and composition, and constant efficiency over a varying load
- disadvantages are related to the additional equipment required for heat exchangers.

Turbines are also available as single- or multiple-shaft machines. The single-shaft machines are simplest with the compressor and energy recovery turbine on the same shaft. This type is best for constant load and speed operation. Multiple-shaft machines have a separate output turbine which is not connected to the compressor or compressor drive turbine. This provides advantages in applications requiring variable load and speed.

5.3.4 Performance

The efficiency of a gas turbine is primarily affected by:

- the type of cycle
- degree of heat recovery
- ambient air temperature
- combustion temperature and pressure
- compressor and turbine efficiencies.

The cycle efficiency for gas turbines is generally expressed in terms of the heat rate (units of BTU/HP-hr) based upon the lower heating value of the fuel. This can be converted to a thermal efficiency by use of a suitable conversion factor (2544 for units of BTU/HP-hr).

One of the biggest advantages of gas turbines is the versatility of their fuel supply. Although gas turbines are basically designed for burning natural gas, alternate liquid and gaseous fuels may be utilized as recommended by the manufacturer. The various fuels available for gas turbines have different characteristics affecting operation, maintenance, and efficiency.

Some fuels available with their characteristics are:

- Natural gas and liquified petroleum gas
 - high purity
 - low emission characteristics
- Fuel oils
 - light distillates
 - heavy residuals
- Solid fuels
 - slurries
- Synthetic gas
 - many properties like natural gas
 - much lower heating value than natural gas.

Due to the lower heating value of some of the alternate fuels, the fuel handling equipment must be sized accordingly to handle the required increase in the fuel flow to maintain the turbine output. In general, all gas turbine fuels require pretreatment

to minimize maintenance and increase the reliability and operational life of the equipment.

The installation of gas turbines requires accommodation of the exhaust gases, the coldest possible air intake for combustion, and space for lubrication/cooling systems and speed/power controllers. A starting system (electric, pneumatic, etc.) will also be required.

5.3.5 Specifications

A gas turbine specification based upon the requirements for the four cases was sent to two manufacturers — General Electric and Westinghouse. Ambient air temperatures of 59°F, sea level pressure, and natural gas fuel were specified (cycle thermal efficiency is based upon the lower heating value of natural gas).

5.3.6 Results

Gas turbines are available to meet driver requirements in Cases 1 through 4. However, no gas turbine was available from the turbine manufacturers in the 2200 HP range for Case 2. Also, for Case 3, a full-size turbine was not available so that a helper steam turbine of about 4000 HP would be required.

Turbine speeds varied widely from 3600 to 9000 rpm, depending on size. Higher powered turbines generally had slower shaft

speeds. For any given turbine, the shaft speed can be varied, but shaft power and efficiencies would be greatly affected.

The thermal efficiencies of gas turbines meeting the specified driver requirements was in the range of 21 - 23 percent without heat recovery. The performance data is shown in Table 5-3.

5.4 DIESEL ENGINES

5.4.1 Introduction

Diesel engines are one of the most efficient types of internal combustion engines and, in addition, a variety of fuels can be used for combustion. Although diesel engines are characterized by high reliability, they do have higher maintenance requirements than steam turbines and electric motors, and somewhat lower reliability. Diesels are characterized by the fact that no external power supplies are required and that they remain efficient at part load operation. Diesel engines are less frequently used to drive very large pumps as the capital and maintenance costs are substantially higher than those of steam turbines or electric motors.

5.4.2 Operation

Diesel motors operate by injecting fuel into the engine cylinder near the conclusion of a compression cycle. The heat generated

DRIVER	DP-1	DP-1	DP-3	DP-3	DP-4	DP-4
CASE NO.	1	1	3	3	4	4
REQUIRED POWER ~ HP	4300	4300	43,000	43,000	21,500	21,500
MANUFACTURER	G.E.	WESTINGHOUSE	WESTINGHOUSE	G.E.	G.E.	WESTINGHOUSE
MODEL	M1502	W41M	CW352	M5332B	M5262A	CW352A
POWER OUTPUT ~ HP	4300	4300	39,800*	33,550*	22,500	22,000
SPEED ~ RPM	8980	8400	4750	4670	3600	4500
EFFICIENCY %	24	20.7	28.2	28.5	26	25.4
DELIVERY PERIOD (weeks)	78	43	61	70	70	61
SIZE (inches) L x W	407 x 104	290 x 96	534 x 130	593 x 124	593 x 124	530 x 130

TABLE 5-3. RESULTS - GAS TURBINES

* Requires use of additional turbine to achieve required pump horsepower.

by the compression is the sole means of igniting the charge. After being ignited, the charge burns and expands, thus converting the heat of the fuel into work.

Diesel engines are available in sizes from a fraction of a horsepower to as large as 48,000 horsepower. Relatively few diesel engines are available in sizes above 16,000 horsepower. Diesel engines are designed to operate at relatively slow speeds (2400 rpm or less). The speed decreases with increased size so that large diesels operate at about 400 rpm. Because of the low speeds of diesels, speed increasing gearing is required for pumping applications.

Diesels can be basically characterized into the following categories:

- Oil-diesel

Air is compressed in the cylinder while the fuel is injected into the cylinder near the conclusion of the compression cycle

- Gas-diesel

A gas mixture is compressed in the cylinder while diesel fuel is injected near the conclusion of the compression cycle, and serves as the ignition source.

These variations in operations significantly affect the engine's performance and economy. Injection of the fuel near the conclusion of the compression cycle permits the use of a higher

compression ratio than obtainable with other internal combustion engines. Higher compression ratios theoretically mean high efficiencies and power (performance).

Diesels can use solid, liquid, and gaseous fuels. The most common fuel is liquid, and the most expensive are high-quality distillate fuels. Lower cost residual fuels can be used. Natural gas is used in gas-diesel engines. In addition, some diesels have been designed to operate on coal dust.

The installation of diesel engines requires a substantial level foundation. The foundation must be insulated against vibration due to the high level of torsional vibration transmitted from the engine. Diesel engine drivers are large; as much as 80 feet is required to accommodate the length of the driver. Consideration will be required in taking appropriate measures to suppress the noise from the air intake and exhaust. Provisions for cooling water (preferably softened water) will also be required.

5.4.3 Performance

The efficiency of diesels is generally expressed in terms of the specific fuel consumption which is defined as:

$$\text{specific fuel consumption} = \frac{\text{lb}_m \text{ fuel/hr}}{\text{HP}_{\text{output}}} \quad (\text{lb/BHP-hr})$$

If the heating value of the fuel is known, the thermal efficiency

of the engine can be determined. The thermal efficiency of a diesel is expressed as:

$$\text{brake thermal efficiency} = \frac{P}{H_F M_F}$$

where P is the shaft output of the engine (BTU/hr), H_F is the heat content (high heat value, HHV) of the fuel (BTU/lb_M), and M_F is the mass of fuel input to the engine (lb_M/hr).

5.4.4 Specifications

Specifications for all four cases were sent to two diesel engine manufacturers — Fairbanks-Morse and Alco. Sea level altitude was specified with 85°F temperature and use of diesel fuel. Engine description and performance information was requested.

5.4.5 Results

Of the various diesel manufacturers contacted, the largest available engine was rated at 27,000 BHP. Although large sizes are available, all but the largest pump drivers in the study are in the range of diesel engines for stationary service. The largest pump driver would require two similar diesel motors operated in tandem. Typical of large diesel motors, the engine's speeds are as low as 400 - 514 rpm. The thermal efficiency of the large diesel motors is as high as 38 percent. A summary of the characteristics of diesel engines for the pumping application is shown in Table 5-4.

DRIVER	DP-1	DP-1	DP-2	DP-2	DP-3	DP-4
CASE NO.	$\xleftrightarrow{1}$ 4300		$\xleftrightarrow{2}$ 2200		3	4
REQUIRED POWER ~ HP					43,000	21,000
MANUFACTURER	F-M*	ALCO	F-M	ALCO	F-M/PIELSTICK	F-M/PIELSTICK
MODEL	COLT PC-2	251F	38TD8-1/8	251C	PC-4	PC-4
# CYLINDERS	V-12	V-18	L-9	V-12	2 x (V-18)	V-18
POWER OUTPUT ~ HP	6000	4400	2520	2250	2 x (27000)	27,000
SPEED ~ RPM	514	1100	720	1100	400	400
THERMAL EFFICIENCY % **	35	35	36	35	35	38
SPECIFIC FUEL CONSUMPTION lbs/BHP-hr	.370	.375	.364	.375	.343	.343
DELIVERY PERIOD (weeks)	104	29	104	29	104	104
SIZE L (inches)	271	247	228	180	1032	516

TABLE 5-4. RESULTS - DIESEL ENGINES

* Fairbanks-Morse

** Based on HHV (High Heat Value) of fuel
of 19,500 BTU/lb.

5.5 DRIVER COUPLINGS

Coupling devices in pump drive systems permit speed matching of the driver and pump, as in the case of a 400 rpm diesel and 3600 rpm pump, or permit variable-speed operation in the case of essentially constant speed devices such as induction motors. Variable speed operation can be economically advantageous where it is necessary to vary the pump operating pressure and flow during the life of the plant.

In diesel motor applications, speed-increasing gears are normally required because of the large speed difference between the pump and driver. A variable drive is not required as the pump speed can be changed by altering the engine speed. For induction motors, a variable speed drive is preferred, as motor speed can only be varied if the frequency of the current is altered. Two types of variable speed drives are eddy-current couplings and fluid couplings. Both of these devices normally will not provide any speed increase function (i.e., output shaft speed will be slower than input shaft speed).

For steam and gas turbines, speed matching devices are often not required as the pump and driver operate at similar speeds; in addition, the driver has a variable operating speed. In cases where speed matching is required, a speed increasing/decreasing gearing would be required.

Constant speed coupling offers high efficiencies, with gears approaching 99 percent in the largest sizes. Variable speed drives are not as efficient, with efficiencies being affected by size and percent of rated load.

In this study, four pump systems of varying power requirements were studied. Within any one of the power ranges, several combinations of pumps/drivers are available. Because of the large number of pump/driver combinations, the pricing information would best be obtained on a case by case basis.

Section 6

ENERGY RECOVERY DEVICES

6.1 ENERGY RECOVERY

In seawater reverse osmosis, considerable energy is used to pump the incoming seawater to high pressures (900 psi in this study) in order to obtain a reasonable flux across the membrane. The fresh water recovered through the membrane is only a fraction of the inlet flow of seawater with current membranes (25 percent and 50 percent was designated in this study). In addition, a relatively low pressure drop exists across the membrane (approximately 50 psi). Thus, a large, high pressure, concentrated brine stream is rejected from the membrane (50 percent to 75 percent of the seawater feed flow in this study) which is available for energy recovery.

Energy can be recovered as mechanical or electrical power from this high pressure stream through hydraulic turbines (including reverse running pumps) and flow work exchangers. For this study, only impulse turbines were considered because of their high efficiency and/or state of development.

The economic feasibility of energy recovery depends upon the seawater flow rate and pressure, the efficiency of the hydraulic device, its capital and operating cost, and its reliability.

The potentials for energy recovery for the four cases studied are shown in Table 6-1 below:

TABLE 6-1. ENERGY RECOVERY OPERATING CONDITIONS

<u>Case</u>	<u>No. 1</u>	<u>No. 2</u>	<u>No. 3</u>	<u>No. 4</u>
Plant Size, (MGD)	2.5	2.5	25	25
Recovery ratio, (%)	25	50	25	50
Feed (gpm)	6940	3470	69400	34700
Reject (gpm)	5210	1730	52100	17300
Hydraulic Power (Hp)*	2200	730	22000	7300

* Based upon 80 percent efficiency and a differential pressure of 900 psi.

6.2 HYDRAULIC TURBINES

Hydraulic turbines are capable of converting the energy in the pressurized brine into mechanical work when reducing a liquid from a higher to a lower pressure. The mechanical work may be used directly or it might be used to drive a generator to obtain electrical power.

There are three types of hydraulic turbines for power recovery systems:

- Pelton wheel, an impulse-type turbine suitable for high heads
- Francis turbine, a reaction-type turbine suitable for medium heads
- Kaplan turbine, a propeller-type turbine suitable for low heads.

The actual operating principles of these three types of turbines are quite different.

6.2.1 Impulse Turbines

For power recovery systems in seawater reverse osmosis plants, impulse-type wheels are suitable for the available high heads. With the impulse-type wheel, the potential energy of the water (in the penstock) is transformed into kinetic energy in a jet issuing from the orifice of a nozzle. This jet discharges freely into the atmosphere against the bowl-shaped buckets of a runner inside a wheel housing. At each revolution the bucket enters, passes through, and passes out of the jet, during which time it receives the full impact force of the jet.

The essential elements of an impulse turbine are:

- a runner
- a jet deflector
- a housing with a guide bearing and shaft.

The runner is basically a disk with integral or attached buckets. Materials such as 13 percent chrome steel or 18-8 stainless steel are advantageous since they have considerably longer life than cast steel and resist pitting caused by cavitation.

The nozzle pipe, together with the needle nozzle and needle tip, forms the jet. The needle is a moving element inside the needle nozzle and needle tip to control the size of the jet impinging on the bucket, thus controlling the power output of the turbine. Because of the high water velocity, erosion of the nozzle and nozzle-valve must be considered in the selection of construction materials such as hard chromium plate.

Hydraulic turbines in power recovery systems driving alternating current electric generators, must have an output shaft speed equivalent to one of the generating synchronous speeds (900, 1800, 3600 rpm for 60 Hz AC generators). The higher turbine speeds are advantageous as higher speed generators are less costly and more efficient.

Progress in hydraulic turbine design has advanced in the area of turbine capacity and power output, but not efficiency as very high efficiencies already exist. An impulse-type turbine has a maximum efficiency of about 90 percent. The maximum efficiency usually occurs at 75 percent of the maximum load. Installation of a hydraulic turbine requires a flume or case to distribute

water as uniformly and smoothly as possible around the circumference of the guide case in order to obtain maximum power and efficiency. Maximum velocity in the turbine and its casing must be limited to ensure steady-state operation. If the turbine is to drive an AC generator, provisions should be made for speed regulation devices.

6.2.2 Impulse Turbine Specification

Four manufacturers of impulse turbines were contacted for price and performance information for the flows shown in Table 6-1 and a head of 2025 feet.

6.2.3 Results

The performance information for the impulse turbines is summarized in Table 6-2. In all cases in this study, an impulse wheel was selected. The estimated power output for the different conditions ranged from 780 to 24,000 Hp. Turbine speeds were between 600 - 1800 rpm. The efficiency of these hydraulic turbines was high; typically in the range of 88 to 90 percent.

6.2.4 Reverse Running Pumps

A centrifugal pump can be modified mechanically and hydraulically for reverse rotation. If modified in this manner, it can be used

CASE NO.	1		2		3		4	
TURBINE	T-1	T-1	T-2	T-2	T-3	T-3	T-4	T-4
MANUFACTURER	A-C*	Krupp	A-C*	Krupp	A-C*	Krupp	A-C*	Krupp
FLOW GPM	5216	5210	1730	1730	52100	52100	17300	17300
POWER OUTPUT (Hp)	2400	2450	780	805	23500	24500	7800	8261
SPEED	1800	1800	1800	1800	600	900	900	1200
PEAK EFFICIENCY %	89.7	88.6	89.7	88.1	89.7	88.6	89.7	90.2
DELIVERY WKS.	—	78	—	78	—	130	—	130

TABLE 6-2. RESULTS - HYDRAULIC TURBINE

* Allis-Chalmers

as an energy recovery device. It can be direct connected (through use of an over-running clutch) to a pump and this can be done in the same casing as for the pump (Ref. 6-1). The characteristics of this method of energy recovery are:

- Cost:

Capital cost for recovery device is slightly greater than for an equivalent centrifugal pump.

- Performance:

The best efficiency is very near that of the equivalent pump.

- Operation:

The device becomes a net consumer of power at about 40 percent or less of the best efficiency point. Thus, partial load operation is not desirable for this type of energy recovery.

These devices are not as efficient as impulse turbines, and are not as efficient at partial load operation. They were not considered further. However, consideration should be given to them for specific plant studies.

6.3 OTHER DEVICES

Devices known as flow-work exchangers have been tested which transfer the pressure energy directly from the brine to the feed streams (Reference 6-1). Operation is shown in Figure 6-1. During the charging process, the feed stream enters the piston chamber and displaces the brine. This operation occurs at low pressure. During the second step, the feed stream is

Figure 6-1
FLOW-WORK EXCHANGER OPERATION

pressurized. The two low pressure open valves are closed, and the high pressure valves are opened. The pressurized brine then pressurizes the feed stream. This process is then repeated.

These devices offer the advantage of high efficiency, particularly in smaller sizes, and potentially lower cost. References 6-2 and 6-3 discuss some of the developments with these devices. These devices were not considered further in this study due to their early stage of development.

Section 7

SYSTEM COMPARISON AND SELECTION

A discussion of the factors involved in the selection of appropriate pumping and energy recovery systems is presented in this section along with a performance and cost summary of the various pumps, drivers, and energy recovery turbines. An example case for an energy recovery turbine selection is given in Section 7.2.2.

7.1 SUMMARY OF CHARACTERISTICS

7.1.1 Pumps

A summary of price information for the four design cases is shown in Table 7-1, and a summary of price and performance is shown in Table 7-2. For pump prices, both a range and average price are shown in Table 7-1. A large range in prices, as supplied by the manufacturers, was obtained for the pumps. The range is attributable to the differences in characteristics of the pumps; e.g., closeness of the fit of the pump to the requirements, degree of sophistication of the specification, and budgetary nature of the price estimate. Individual prices, as offered by the manufacturers, are not given for this reason.

The unit prices for the pumps (Table 7-2) are given in terms of feed flow and horsepower. It is seen that there is an economy

CASE NO.	PUMP NO.	PLANT SIZE (MGD)	RECOVERY RATIO %	PUMP FLOW (GPM)	AVERAGE PRICE (Thou. \$)	PRICE RANGE (Thou. \$)	NO. OF QUOTES
1	P-1	2.5	25	6940	168	70 - 350	6
2	P-2	2.5	50	3470	86	65 - 125	5
3	P-3	25	25	69,400	1,023	975 - 1,070	2
4	P-4	25	50	34,700	523	350 - 915	4

TABLE 7-1. PUMP PRICE INFORMATION

DESCRIPTION	2.5 MGD PLANT	25 MGD PLANT
Efficiency (%)	81 - 87	83 - 89
Pump Speed (rpm)	3560 - 7650	1780 - 4450
Unit Price: (\$/gpm of feed)	24 - 25	15
(\$/Hp)*	38 - 40	25
Delivery Time (Weeks)	32 - 104	40 - 104
Reference Section	4.2	4.2

TABLE 7-2. PUMP SUMMARY - 2.5 and 25 MGD PLANT

*900 psi differential head.

of scale due to pump size. There is also an increase in pump efficiency with size. However, there is a large variation within a given pump size depending upon the closeness of fit of pump to the design operating point. This variation in efficiency should be taken into account in a final driver selection since, in most cases, the energy cost is more significant than the capital cost (see Section 7.2).

Pump delivery times of up to two years may be expected. This could significantly affect the construction schedule of the RO plant.

7.1.2 Drivers

A summary of the price and performance information of the four types of drivers is shown in Table 7-3. Information is shown both by plant size and recovery ratio. Efficiency is shown for each device in terms of its customary efficiency rating since cost of fuel and operation are generally calculated based upon these efficiencies. (See referenced sections for discussion of these efficiencies.) It should be noted these efficiencies are not directly comparable to each other due to differences in their basis of calculation.

Cost information is shown for the driver only without any support equipment. In some cases, this support equipment can be more

DESCRIPTION	PLANT SIZE (MGD)	DRIVER			
		Electric Motor	Diesel	Gas Turbine	Steam Turbine
Efficiency (%)	2.5 ↓	95 - 96	35 - 37	21 - 27	56 - 69
Efficiency Type		Engine	Thermal	Thermal	Engine
Price (Thou. \$):					
@ 25% recovery		83	780	1100	596
@ 50% recovery		55	350	N/A*	286
Unit Price (\$/Hp)		19 - 25	160 - 180	260	130 - 140
Delivery (weeks)		—	29 - 104	43 - 78	52 - 75
Efficiency (%)	25 ↓	95 - 96	38	23 - 28	71 - 74
Price (Thou. \$):					
@ 25% recovery		N/A	12,300	2,900**	1,120
@ 50% recovery		280	6,500	2,800	927
Unit Price (\$/Hp)		13	290 - 300	75 - 130	26 - 43
Delivery (weeks)		—	104	60 - 70	60 - 75
Reference Section	2.5 & 25	5.1	5.4	5.3	5.2
Variable Speed		No	Yes	Yes	Yes
Source of Energy		Electricity	Various Fossil Fuels	Various Fossil Fuels	Steam

TABLE 7-3. PUMP DRIVER SUMMARY - 2.5 and 25 MGD

*N/A - Not available.

**Requires helper turbine.

costly than the driver itself. For example, the electric motor requires a source of electricity. If this is not available in sufficient size at the site, then electrical transmission lines and substations must be provided. Similarly, a source of steam and cooling water is required for the steam turbine. In addition, the cost and availability of energy can be very site specific. For example, in some parts of the Middle East natural gas is presently flared, although it could be used as a source of fuel for the RO plant. Thus, a comparison of the various driver alternatives should be made based upon site specific information.

Delivery times for the drivers is similar to that of the pumps in that times of up to two years can be expected. Thus, these too could significantly affect the construction time for the plant.

7.1.3 Energy Recovery Turbines

The price and performance summary for the energy recovery turbines is shown in Table 7-4. From this information, it can be seen that:

- the efficiency of impulse type of energy recovery turbines is very high
- the price in the smaller sizes is nearly independent of size

DESCRIPTION	2.5 MGD PLANT	25 MGD PLANT
Price (Thou. \$):		
@ 25% recovery	280	1200
@ 50% recovery	225	625
Unit Price (\$/Hp)	115 - 320	47 - 85
Efficiency (%)	88 - 90	89 - 90
Delivery Time (Wks.)	80	130
Speed (rpm)	1800	600 - 1200
Reference Section	6.2	6.2

TABLE 7-4. ENERGY RECOVERY TURBINE SUMMARY -
2.5 and 25 MGD PLANT

- delivery times can be even longer than that of the pumps and drivers.

This can be explained by the fact that there is a great deal of custom engineering in the impulse turbines. Therefore, in the smaller size applications, consideration should be given to use of reverse running pumps which do not require the same degree of detailed engineering.

7.2 SYSTEM SELECTION

Shown in Figure 7-1 is the cost of product water for a 25 MGD seawater RO plant (Ref. 7-1). Shown are the capital components for the pump and energy recovery system and the power cost of various electrical power rates. For this particular system, spare pumps are provided to assure a 90 percent capacity factor, and energy recovery turbines are used to generate electrical power. It can be seen that:

- the cost of the electrical energy is generally much more than the annualized pumping system equipment cost
- in low energy cost areas, energy recovery may not be worthwhile
- high equipment efficiency is necessary in high energy cost areas
- the annual power cost is large enough to justify a detailed study of the pumping system (e.g., power cost of as much as \$12 million per year for a 25 MGD plant).

Figure 7-1
SEAWATER RO COST OF PRODUCT WATER

7.2.1 Pumps and Drivers

To select a pump and driver system requires an evaluation of many parameters, most of which are site specific. These factors can be classified as either tangible or intangible, and an example of some of these factors follows:

- Tangible:

- Cost of Capital
- Energy Cost and Availability
- Component Efficiency
- Type of Driver
- Plant Loading
- Overall Plant Reliability

- Intangible:

- Start-up Requirements
- Operation and Maintenance Requirements
- Simplicity

7.2.2 Energy Recovery

The desirability of installing an energy recovery system and the contribution of it to the overall water cost is determined by the capital cost, operating and maintenance cost, and energy savings of the energy recovery system.

Capital Cost

Based upon the product flow and recovery ratio required, the system feed rate can be determined. Pressure requirements will

further specify the pump system. Based upon site specific conditions, the appropriate driver can be selected. The driver should be able to handle the energy pumping horsepower requirement, independent of eventual energy recovery considerations. This is desirable for start-up and in case of failure of the energy recovery system. Energy recovery, therefore, will not reduce the capital cost for the pump and drivers.

The horsepower requirements to be delivered by the pump driver are given by the formula:

$$Hp = \frac{\Delta P \times Q}{1714 \times \eta \times y}$$

where:

Hp = Energy to be delivered by the pump driver (Hp)

ΔP = Pressure increase in pump (psi)

Q = Produced flowrate (gpm)

η = Overall efficiency from driver through pump (including coupling efficiency)

y = Water recovery ratio = $\frac{\text{Product Flow}}{\text{Feed Flow}}$

If an energy recovery unit is installed, the energy contained in the high pressure waste brine can be utilized for the high pressure pump. The energy saved and supplied to the pump is given by the formula:

$$Hp^1 = \frac{\Delta P^1 \times Q}{1714 \times \eta^1 (1 - y)}$$

where:

H_p^1 = Energy supplied to the high pressure pump by the energy recovery unit (H_p)

η^1 = Overall efficiency between energy recovery turbine and high pressure pump (not including pump efficiency)

ΔP^1 = Pressure drop in energy recovery turbine (psi).

During continuous operation, when utilizing an energy recovery turbine, the net energy to be delivered to the pump driver is given by:

$$H_{p,net} = \frac{Q}{1714 \times \eta} \left[\frac{\Delta P}{y} - \frac{\Delta P^1}{\eta^1 (1 - y)} \right]$$

The installed capital cost is determined by:

- equipment cost
- installation labor and materials
- indirect field costs
- engineering services
- contingency
- interest during construction
- operator training and startup.

The annual capital cost can then be determined, based on depreciation, return on investment, insurance and taxes to arrive at a capital recovery factor. With a given plant capacity factor, the annual capital cost per thousand gallons of product water

can then be calculated by:

$$\text{Water Cost (\$/1000 gal.)} = \frac{\text{Capital Cost} \times \text{CRF}}{Q^1 \times \text{CF} \times 365}$$

where:

CRF = Capital Recovery Factor

CF = Capacity Factor

Q^1 = Daily Product Flow (1000's gallons/day)

Operating and Maintenance Cost

The contributions to the operating and maintenance cost are the energy cost, the labor requirements for operation and maintenance, and the requirements for supplies and materials.

The energy cost, in terms of electrical energy, is given by:

$$\text{Energy Cost} = \frac{\text{EC} \times \text{Hp}_{\text{net}} \times 0.746 \times 24}{Q^1}$$

where:

EC = Electricity Cost (\$/Kw-Hr)

Energy Cost (\$/1000 gallons of product)

Example Case

A separate economic analysis needs to be conducted to determine the desirability of installing an energy recovery system as determined by:

- the cost of the energy recovery unit
- total energy to be recovered (as function of prime pressure and recovery ratio)
- cost of energy.

An example case for determining the necessity of using energy recovery is shown in Figure 7-2. The following factors were used in the analysis:

- Brine pressure (850 psi)
- Recovery ratio (25 and 50 percent)
- Product flow rate (2.5 MGD)
- Cost of energy recovery unit (from Table 6-2)
- Capital recovery factor (12.5 percent)
- Electrical energy cost (variable)
- Operation and maintenance cost (10 percent of capital cost).

For this particular case, the "break-even" point varies between 0.023 and 0.033 \$/Kw-Hr for 25 and 50 percent recovery ratio plants. At power costs below these values, energy recovery is not worthwhile, while above this rate, energy recovery should be considered. Larger size plants can be expected to show "break-even" points at even lower power costs due to the lower unit costs for larger size energy recovery turbines.

Figure 7-2

ENERGY RECOVERY SYSTEM
INCREMENTAL COST COMPARISON

CASE A

2.5 MGD
25% RECOVERY RATIO
POWER SAVED = 1080 KW-HR/HR

CASE B

2.5 MGD
50% RECOVERY RATIO
POWER SAVED = 540 KW-HR/HR

APPENDIX

List of Manufacturers Providing Information Used in This Report

Pumps

Mr. Edy Sennhauser
Sulzer Bros.
1255 Post Street, Suite 911
San Francisco, CA 94109

Mr. Fred Alban
Ingersoll-Rand
101 Howard Street, 42A
San Francisco, CA 94105

Mr. David Seabury
Bingham-Willamette Co.
P. O. Box 878
Orinda, CA

Mr. Tom Arnold
Worthington Pumps
324 Hearst Building
San Francisco, CA 94103

Mr. Larry Greaves
United Pumps
3557 Mt. Diablo Blvd.
Lafayette, CA

Diesel Engines

Mr. B. N. Ghinazzi
Fairbanks-Morse
1485 Bayshore Blvd.
P. O. Box 63
San Francisco, CA 94124

Mr. E. T. Flagg
Engine Power Co./ALCO
P. O. Box 8130
Stockton, CA 95207

Steam Turbines

Mr. Robert J. Horvath
General Electric Co.
55 Hawthorne Street
San Francisco, CA 94105

Mr. Larry Pearson
Elliott Turbines
P. O. Box 699
Orinda, CA 94563

Gas Turbines

Mr. Robert J. Horvath
General Electric Co.
55 Hawthorne Street
San Francisco, CA 94105

Mr. Francis Taylor
Westinghouse Electric Corp.
One Maritime Plaza
San Francisco, CA 94119

AC Motors

Mr. Glen Wimer
General Electric Co.
P. O. Box 3736
San Francisco, CA 94119

Mr. Francis Taylor
Westinghouse Electric Corp.
One Maritime Plaza
San Francisco, CA 94119

Hydraulic Turbines

Mr. Wolfgang Horn
Krupp International, Inc.
582 Market Street
San Francisco, CA 94104

Mr. Ron Miller
Allis-Chalmers Corp.
Hydro-Turbine Division
East Berline Road
P. O. Box 712
York, PA 17405

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