Advanced High Efficiency Energy Recovery

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Abstract

In a paper[1] presented at the IDA World Congress in Bahrain in 2002, an evaluation was undertaken of the commercial, major energy recovery devices (ERD). These units included the impulse turbine (Pelton wheel), hydraulic turbo charger, work exchanger, pressure exchanger and reverse running pump (Francis turbine). It was found the most energy efficient systems were the two exchangers; the least plausible device was the reverse running pump. The Francis turbine is installed in a number of older membrane desalination plants, for it was the first such unit offered to the trade.

A detailed examination of the cases depicted in the paper indicated that, for large plants, the impulse turbine was now closely approximating its highest practical, operational efficiency, 90%. The two exchangers, with pressure transfer efficiencies of 92% - 94%, have little room for further improvement except for reducing the level of fluid being bypassed from the brine to feed streams. Bypassing now amounts to about 3% to 6%. Demonstration of equipment life and scale-up of the pressure exchanger beyond small and medium size plants is required. The work exchanger has favorable economic value, basically, in plants whose train sizes are in excess of about 6,000 m^3/d (1.6 MGD).

The most intriguing ERD, is the one having the largest potential for minimizing energy consumption, the new AT line of hydraulic turbochargers. The manufacturer of the AT hydraulic turbocharger, Pump Engineering, Inc., has revamped its production system to incorporate the latest 3D CAD (Computer Aided Design) and CAM (Computer Aided Manufacturing) software, resulting in high efficiency, complex geometry impellers. The turbine rotors and pump impellers are now produced on 5 axis CNC milling machines. These changes have resulted in a significant increase in the hydraulic energy transfer efficiency and a corresponding decrease in SWRO energy consumption to industry best levels. This advanced turbocharger also makes possible, in train sizes greater than 3,750 m^3/d (1 MGD), the use of highly efficient and cost effective single stage pumps for high-pressure duty. The product line, with its small footprint, is suitable for the large size trains currently being designed for mega plants.

This paper discusses this newly optimized ERD as it is now being applied in commercial installations. Emphasis is on plant net energy rate and Total Life Cycle Cost of the high-pressure pump and ERD.

I. INTRODUCTION

In determining the total cost of water for any membrane water treatment plant, whether it is brackish or sea water, the most dominant factors are the net energy consumption of the facility and the amortization of the plant over a set time period at a given interest rate. These two items account for 75% to 80% of the cost. Membrane replacements, maintenance, chemicals, labor and supervision etc., each, is a small percentage of the water costs. It is for this reason that in the last few years water treatment plant suppliers have looked at developments that could lower energy and amortizations costs. The latter item has given rise to many creative financial packages as well as innovative technology that could lower system capital costs.

The energy consumption line item, however, is a function of membrane technology and the ability to recover and reuse the energy from the reject stream. In recent years, membrane manufacturers have made notable progress in improving the functionality of membranes, all aimed at lowering the system applied pressure. Energy recovery has been successfully employed, for decades now, in seawater systems where the brine reject stream has a high flow and substantial pressure. The need is now to improve the efficiency of these energy recovery devices (ERD) and to increase their capacity so as to be able to process, economically, the flows from the mega size plants now coming on-stream and on the drawing boards.

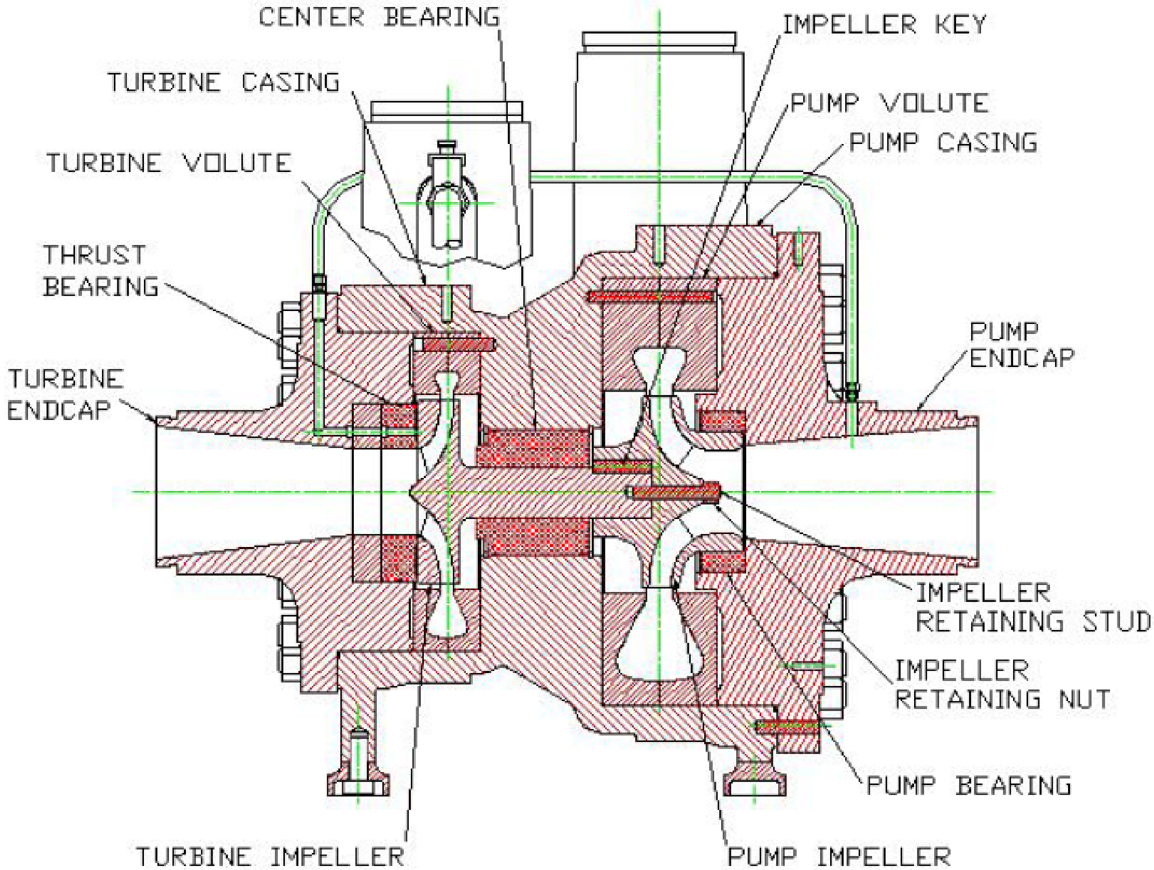
A large capacity, more efficient high-pressure ERD, turbocharger, has been developed by Pump Engineering, Inc. using a computer-derived design that employs high efficiency hydraulic designs and materials of construction appropriate for high salinity and pressure applications. This newly designed turbocharger can recover up to 82% of the waste brine stream energy.

This paper describes this high pressure turbocharger (AT Turbo) and its application in several seawater desalination RO plants. Data and energy consumption are provided. Also, applications of the AT Turbo in related water treatment plants are provided.

II. THE AT TURBOCHARGER

Figure 1 is a cross section of the newly redeveloped AT-4800 turbocharger and Figure 2 is a photograph of the same unit. The AT Turbo external pressure housings consist of a center casing, containing pump and turbine sections and separate pump and turbine end caps. The center casing serves as pressure containment for removable volute inserts for both pump and turbine sections. The pump and turbine end caps have precision fits with the center casing and are secured to it by means of bolting studs and nuts. The turbocharger rotor consists of pump impeller and turbine runner. The pump impeller is driven by the turbine shaft through a key. It is retained on the turbine shaft by the impeller stud and nut. The rotating assembly is radially positioned by center and pump bearings. The rotor, itself, is positioned axially by a hydrostatic thrust bearing. All bearings are mounted on O-rings. The center and pump bearings are positioned by spiral retaining rings. All bearings are made of resin impregnated carbon graphite grade M596 or optional alumna oxide. The rotating elements are constructed of super duplex stainless steel alloy 2507. The casing components are offered in duplex stainless alloy 2205 and super duplex 2507. An auxiliary line and valve are provided to a secondary nozzle to vary brine pressure and flow in a 20% – 25% range for membrane plant control. The auxiliary nozzle directs flow into the turbine volute 90^0 from the main nozzle and, since it is controlled by a valve, acts as a variable area nozzle. The auxiliary nozzle, then, is able to change brine pressure and flow rate without throttling or bypassing.

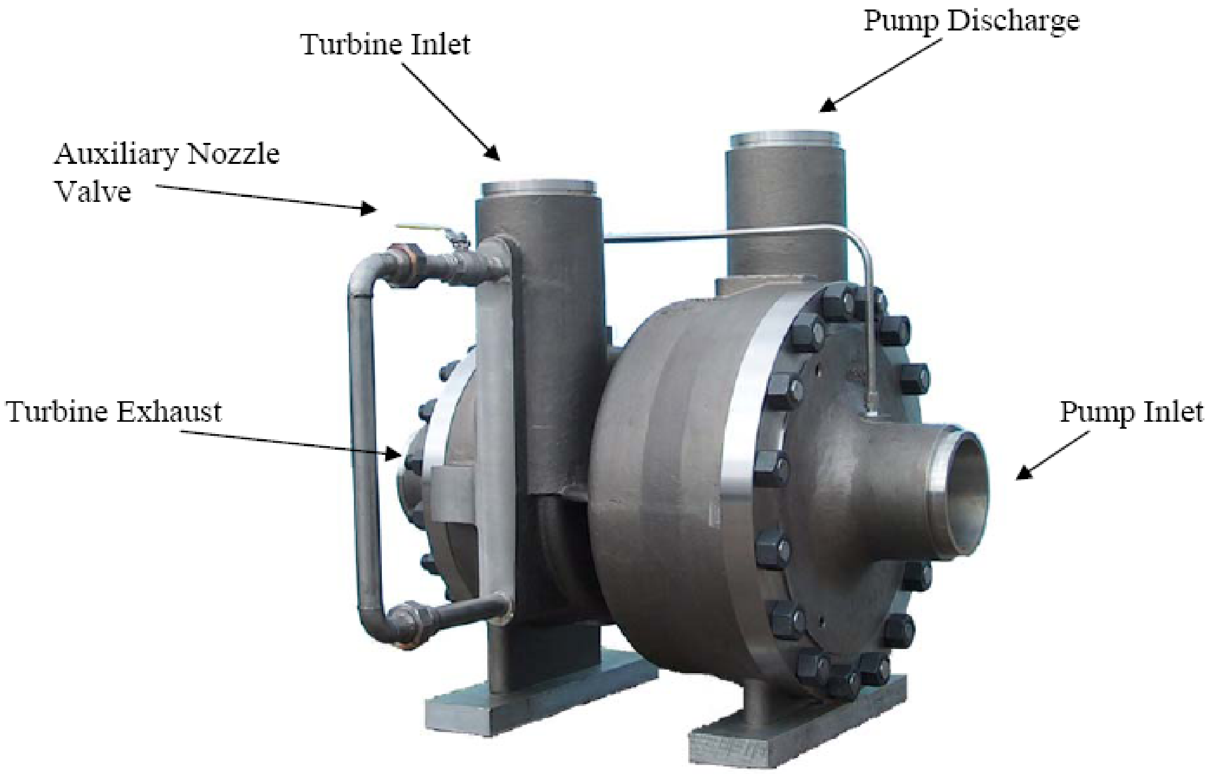
Figure 1
Cross Section of HTC AT-4800



III. DESIGN FEATURES

The AT Turbo is an integral turbine driven centrifugal pump. The turbine is a single stage radial inflow type. The pump is a single stage centrifugal type with its impeller mounted on the turbine shaft. The unit is entirely energized by the high pressure brine stream. The AT design has been optimized for simplicity of application and operation, efficiency, reliability and maintainability. The AT Turbo's high speed operation (large units operate at 5,000rpm – 8,000rpm) allows the use of single stage impellers for a simple end suction/discharge design, avoiding the complexity and cost of multistage construction. The end cap design also allows for easy inspection and maintenance of the bearings and rotor. As can be seen in Figure 1 cross section, the AT Turbo's rotor is entirely contained within the casing thereby eliminating the need for costly and maintenance intensive mechanical shaft seals. Likewise, the use of water lubricated bearings eliminates the need for oil lubrication and the auxiliary oil lubricating equipment. The most notable feature of the AT turbo design, however, is the US and International patent pending Volute Inserts design. The Volute Inserts are radially split mirror image halves that allow the complete machining and surface finishing of the volute's waterway. This design makes possible the custom manufacturing of a completely original and optimal hydraulic design for every AT Turbo order. No longer will pump users have to accept off design operation when the cast volute of the pump does not match the flow duty point. Also, the Volute Insert design allows for an economical re-rate of the Turbo. This feature is especially useful if either feed water quality or membrane performance changes significantly from design projections. The Volute Insert could be remachined to match new conditions or new inserts could be manufactured for a fraction of the cost of a new unit. Either scenario would be preferable to a lifetime of energy wasting off design performance.

Figure 2
Photograph of HTC AT-4800



IV. HIGH EFFICIENCY DESIGN AND MANUFACTURING PROCESS

4.1. Engineering and Design

The AT Turbo development has resulted in efficiency gains of up to 40% over the performance of first generation turbochargers. The key to the performance improvements is the development of an agile engineering, design, and manufacturing process that allows the creation of a totally original and optimized hydraulic flow path for each individual project. Thus, every AT Turbo is designed to operate at the highest efficiency at the customer's unique operating conditions.

The goal of the design process is to match the turbine power output with the pump impeller power absorption, with both occurring at the same speed and at their respective best efficiency points. This is a complex task and can only be accomplished by use of advance computer based design methods combined with years of data collection of experimental and production testing.

The process starts with the confirmation of the customer's operating data. Such data would include maximum and minimum membrane pressures and flow envelope as well as the recovery ratio range. The first step in the design process is to select the optimal running rpm. The rpm selection will determine the specific speed, hence impeller meridonal profile. Factors to consider in speed selection are diameter vs shroud friction losses, bearing limitations, and resultant shroud thrust loads.

Turbo machine design software is used to create a meridonal profile shape for both pump impeller and turbine runner. Number of blades for turbine runner and pump impeller are selected for optimal torque transfer. The number of blades tends to be higher for the turbine side. Blade leading and trailing edges are shaped to minimize entrance and exit shock losses. Figures 3 and 4 illustrate this part of the process. The design software provides a number of numerical and graphic outputs that allow the designer to compare and contrast design changes in the optimization process.

Figure 3
Turbine Impeller

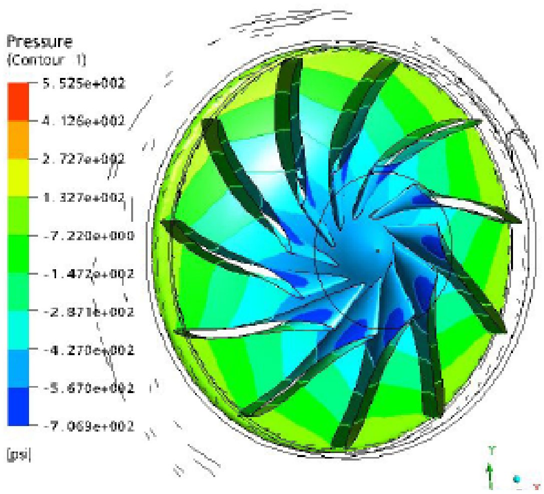
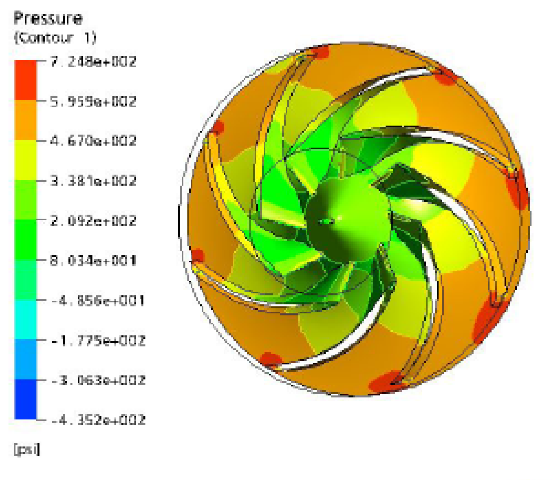


Figure 4
Pump Impeller



	Pump		**Turbine**	
Mass Flow Rate	805.07	Lbm/s	490.528	Lbm/s
Volume Flow Rate	12.5204	Ft^3/s	7.62865	Ft^3/s
Total Blade Torque	1434.27	Ft-lbf	-1701.19	Ft-lbf
Headrise	1110.12	Ft	-2256.71	Ft
Power Input	1242.2	kW		kW
Inlet Flow Coefficient	0.347558		0.393331	
Exit Flow Coefficient	0.171268		0.603523	
Head Coefficient (LE-TE)	0.085288		-1.43055	
Total Efficiency (LE-TE)	0.953741		0.978458	
Static Efficiency (LE-TE)	0.82635		0.583816	
Static Headrise	961.844	ft		

Table 1
HTC-4800 Pump and Impeller Design Parameters

Figure 5
Turbine Fluid Flow

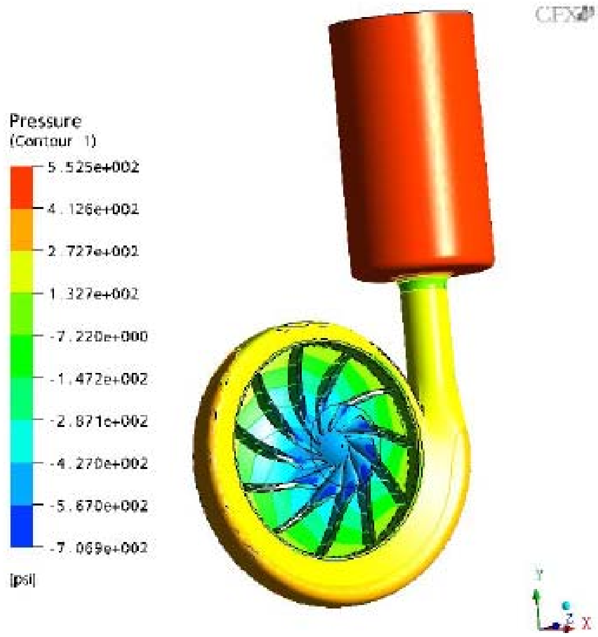
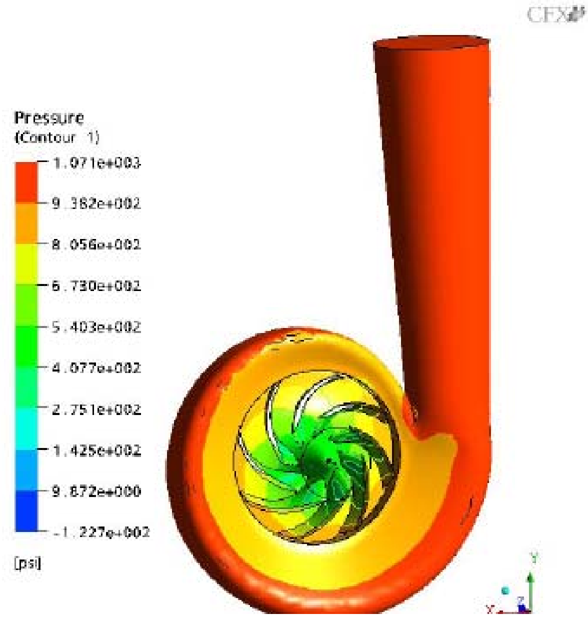


Figure 6
Pump Fluid Flow



Once the impeller and runner geometry have been produced, the stationary hydraulic path, consisting of the turbine and pump volutes, pump diffuser, and turbine nozzles components are designed (Table 1). They are then connected to the impeller and meshed into a single flow path that is subjected to a fluid dynamic finite element analysis using Ansis CFX–5 software. Figures 5 and 6 show graphic displays of turbine and pump fluid flows, while Table 1 shows numeric data output.

4.2. Manufacturing

With the completion of engineering and design work, an IGES file is generated by the design software and exported to a 3D CAD and 3D CAM software. The CAD software is used to generate 2D and solid drawings of the flow path elements. The CAM software generates the tool path, Figure 8 and CNC machining code for the machining of impeller blade geometry on 5 axis CNC milling machines. Figure 7 is a 3D CAD drawing of a turbine runner, while Figure 9 shows an impeller being machined on a 5 axis CNC milling machine. Figure 10 shows the 3 axis machining of a volute insert. The manufacturing of the casings and end caps are performed with conventional vertical turret lathes and CNC 3 axis milling machines.

V. AT TURBO PERFORMANCE

Generally, an ERD is rated as having a certain efficiency based on the conversion of hydraulic energy into mechanical shaft energy. However, in reverse osmosis (RO) where the process is driven by pressure energy, the shaft energy generated by the ERD is normally transferred to the feed pump, which, then, converts that energy back into pressure energy in the feed stream. The most accurate measure of ERD efficiency for an RO system is the ratio of hydraulic energy returned to the feed stream to the amount available in the brine stream. This ratio is called the Hydraulic Transfer Efficiency (*n*te) and is defined as:

$$nte = H_{out} / H_{in} \qquad [\,1\,]$$

Where:

H_{out} = Hydraulic energy transferred to the feed stream
H_{in} = Hydraulic energy available in the brine stream

Figure 7
2D Turbine Impeller

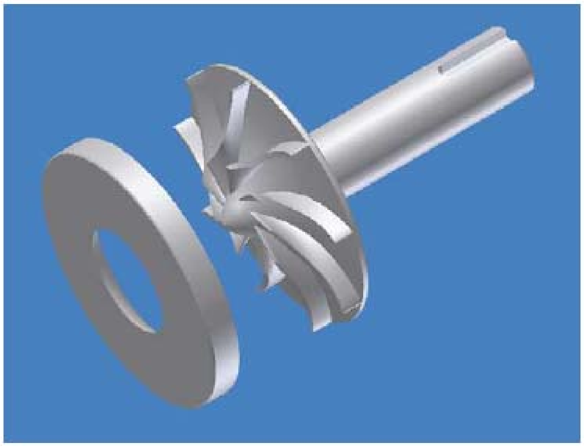


Figure 8
CAM Path

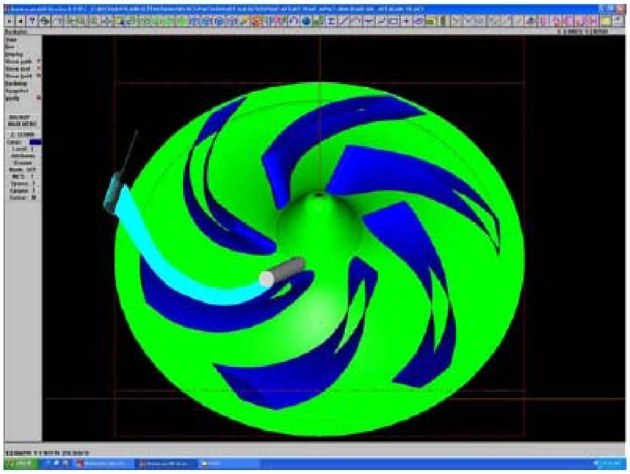


Figure 9
3D Impeller Machining

Figure 10
Volute Insert Machining

Hydraulic transfer efficiency is the truest method of evaluating and comparing the effectiveness of any energy recovery device, including impulse turbines, flow work exchangers, and Hydraulic Turbochargers. Unlike conventional ERDs, such as reverse running pump turbines and Pelton impulse turbines, the energy transfer efficiency of the AT Turbo is independent of pump efficiency. The reason for this is the Turbo contains its own pump, so the complete energy transference occurs within the Turbo. The rotor speed is completely independent of the motor/high-pressure pump. Thus, the AT Turbo can be designed for high speed operation, which is the most efficient and cost effective design. Pump efficiencies of 90%+ are possible for larger AT Turbos.

The useful work of the AT Turbo is expressed as the "Boost Pressure", ΔP. This is the pressure rise that occurs between the Turbo's pump inlet and pump discharge. To apply the Turbo to an RO system, the boost pressure needs to be calculated. Figure 11 is used to find the approximate transfer efficiency ***n*te** for the Turbo. For example, at a feed flow rate of 6,000gpm, the AT Turbo displays an *n*te of about 80% - 81%. Determining the ***n*te** makes calculation of the Turbo pressure boost, **ΔP**, straight forward:

$$\Delta P = (nte)\,(Rr)\,(Pbr - Pe) \qquad [\,2\,]$$

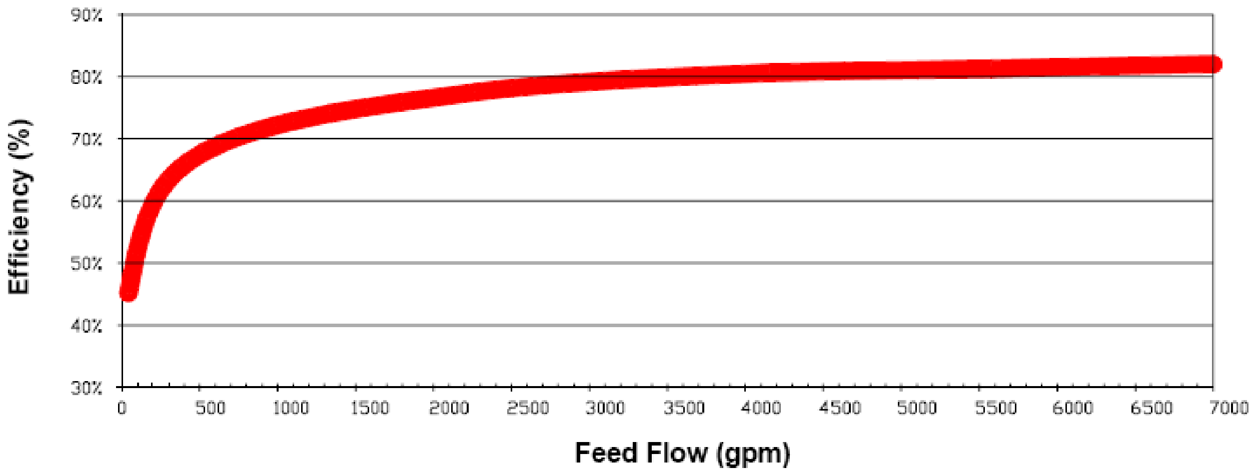
Where:

Rr = ratio of brine flow to feed flow
Pbr = brine pressure at turbine inlet
Pe = exhaust pressure of Turbo (brine pressure leaving Turbo)

$$\Delta Pbr = (Pbr - Pe) \qquad [\,3\,]$$

Substituting data into the formula for the 6,000 gpm (1,363 m^3/h) example is as follows:
Qf = 6,000 gpm (feed flow)
Qb = 3,600 gpm (brine flow)
Qp = 2,400 gpm (product flow)
y = 40% (recovery ratio)
Rr = 60% (reject ratio)
Pm = 1,000 psi (membrane pressure)
Pbr = 980 psi (brine pressure)
Pe = 5psi (brine exhaust pressure)
ΔP = (0 .80) (3600gpm/6000gpm) (980psi – 5psi) = 468 psi

Figure 11
AT Turbo Hydraulic Energy Transfer Efficiency (nte)



A design objective is to make ΔP as large as possible to obtain maximum energy recovery. Equations 2 and 3 indicate that increasing ΔPbr will accomplish this. Since Pbr is usually a given value based on membrane design, the turbine exhaust pressure should be at a minimum safe level. Although the AT Turbo can discharge brine at a high pressure, this exhaust pressure does reduce the available recoverable energy.

In the above example, the use of the AT Turbo reduced the high pressure feed pump discharge pressure from 1000 psi to 532 psi. Thus, not only will the AT Turbo system be energy efficient, but it will contribute capital cost savings as well, as the pump will be smaller than if the Turbo was not employed. For instance, assuming a suction pressure of 50 psi to the high pressure pump, high pressure pump pressure differential is 482 psi. For flows of this magnitude, a single stage centrifugal pump displaying 88% efficiency is available. The single stage pump will be less costly from both a capital and maintenance basis.

Please see Table 2 for actual test data from an HTC AT-1800.

Table 2

CERTIFIED PERFORMANCE TEST

PEI TEST REPORT	**CUSTOMER:**	ITT WET			
	S.N.:	2202C	**J.N.:**	C24059-1	**MODEL:** HTC AT-1800
	DATE:	9/27/2004	**TEST:**	TA-2474	**NOZZLE:** 1.5

DATA AUX CLOSED		**DATA AUX OPEN**		**DESIGN COND.**	
PFDIS	165.2	PFDIS	152.2	Q FEED	1989
PFIN	105.3	PFIN	101.2	S.G.	1.03
PTDIS	38.48	PTDIS	38.66	Q BRINE	1292.9
PTIN	159.9	PTIN	146.7	MEM P.	906
QB	489.6	QB	490	T IN. P.	888
QP	250.6	QP	262.1	T OUT. P.	5
				FEED IN P.	481.8

PERFORMANCE RESULTS	Aux Closed	Aux Open
Projected Delta P Turbine *	872.12	774.75
Efficiency (%) (Test)	74.58	72.45
Boost (Test)	59.90	51.00
Feed Flow (Test)	740.20	752.10
Differential Pressure Brine (Test)	121.42	108.04
Reject Ratio (%) (Test)	66.14	65.15

VI. APPLICATIONS EMPLOYING THE AT TURBO

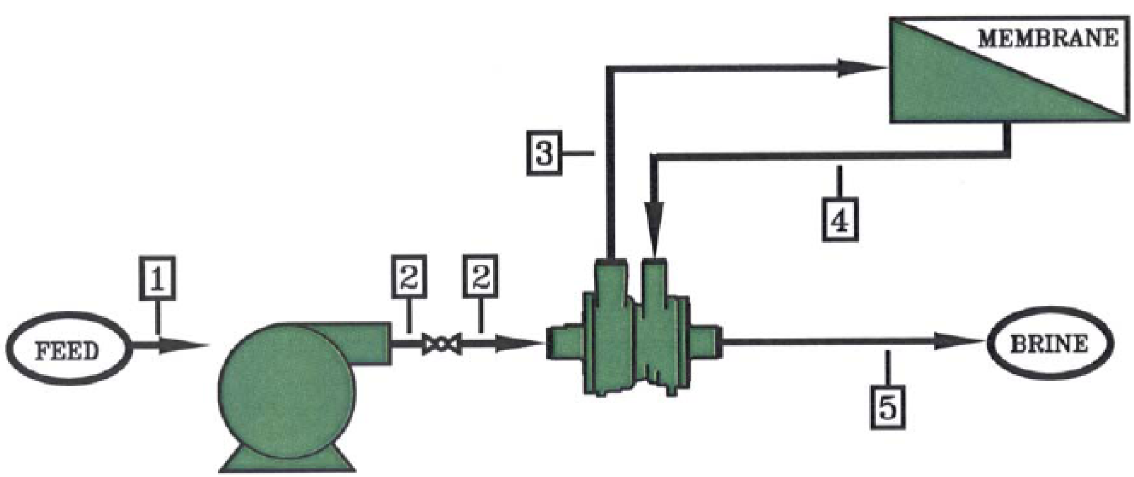
Four SWRO cases are discussed for specific energy conservation. The first example has the following conditions and is illustrated in Figure 12. (Note: The first two examples are based on AT Turbo units constructed in 2004.)

6.1. Example 1 - HTC AT-1800

Feed Pump Efficiency: 82%, Motor Efficiency: 95%, HTC AT-1800 *n*te: 74%

Point	Flow (gpm)	Pressure (psi)	Flow (m3/h)	Pressure (Bar)
1	1960	40	445.5	2.76
2	1960	486	445.5	33.5
3	1960	890	445.5	61.3
4	1176	870	267	60
5	1176	5	267	.34

Figure 12
Flow Schematic



The Turbo Boost Pressure for this example is calculated using Equation 2 on page 8.

ΔP = (*n*te) (Rr) (Pbr – Pe)

ΔP = (0.74) (0.60) (865 psi) = **384 psi**

The main feed pump differential pressure (assuming 40 psi suction pressure) is:

890 psi – 384 psi – 40 psi = 466 psi

The main feed pump power consumption is determined by the equation:

Q(gpm) x P(psi) x (0.0005831)/*n* = WHP(water horsepower) or

1,960 x 466 x 0.0005831 = 532.7/.82 = 649.5 shaft horsepower

649.5 shaft horsepower x 0.746 = 484.3kW/0.96 = 504.5kW wire to water power

Specific Energy Rate for the high pressure circuit = kWh/1,000gal permeate production

504.5kW/47.04 = 10.72kWh/1,000gal or **2.83kWh/m3**

6.2. Example 2. - Projections Based on SWRO plant on Arabian Gulf

Feed Pump Efficiency: 87% Motor Efficiency: 96% HTC AT-4800 *n*te: 80.5%

Point	Flow (gpm)	Pressure (psi)	Flow (m3/h)	Pressure (Bar)
1	6157	40	1399	2.76
2	6157	568	1399	39.1
3	6157	1075	1399	61.3
4	3694	1050	839	60
5	3694	10	839	.69

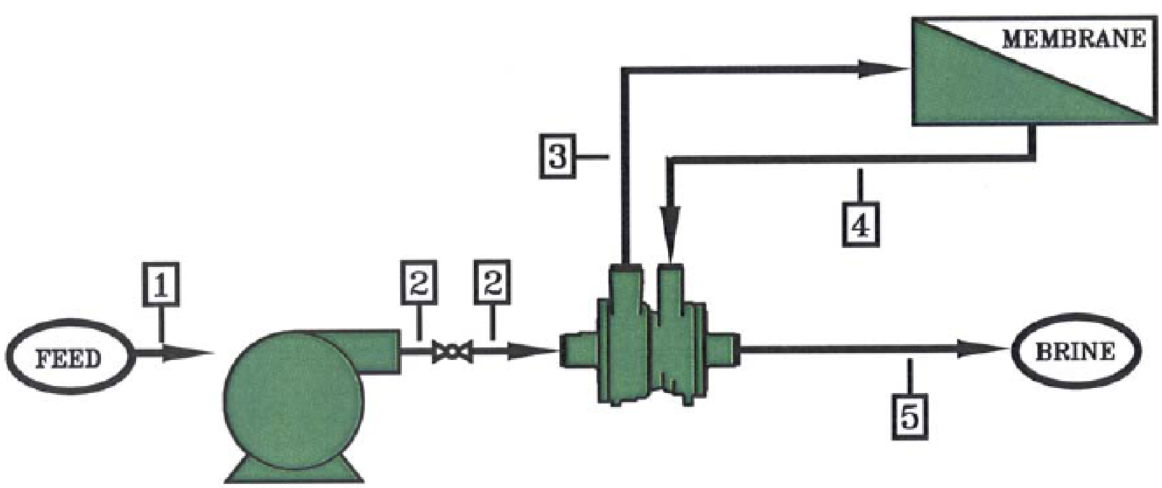


Figure 13
Flow Schematic

The Turbo Boost Pressure for this example is calculated using Equation 2 on page 8.

$$\Delta P = (nte)\,(Rr)\,(Pbr - Pe)$$

$$\Delta P = (0.805)\,(0.60)\,(1040 \text{ psi}) = \mathbf{502 \text{ psi}}$$

The high pressure pump differential (assuming 40 psi suction pressure) is:

1,075 psi – 502 psi – 40psi = 533 psi

Main feed pump power consumption is determined by the equation:

Q(gpm) x P(psi) x (0.0005831)/*n* = WHP(water horsepower) or

6,157 x 533 x 0.0005831 = 1913.5/0.88 = 2,174 shaft horsepower

2,174 shaft horsepower x 0.746 = 1621 kW/0.96 = 1689 kW wire to water power

Specific Energy Rate for the high pressure circuit = kWh/1,000gal permeate production

1689 kW/147.7 = 11.43 kWh/1,000gal or **3.02 kWh/m3**

6.3. Theoretical Projections Based on Mediterranean Sea SWRO Plant

Feed Pump Efficiency: 92% Motor Efficiency: 96% HTC AT-9600 *n*te: 83%

Point	Flow (gpm)	Pressure (psi)	Flow (m3/h)	Pressure (Bar)
1	11,500	40	2612	2.76
2	11,500	422	2612	29.1
3	11,500	820	2612	56.55
4	6900	800	1567	55
5	6900	5	1567	.34

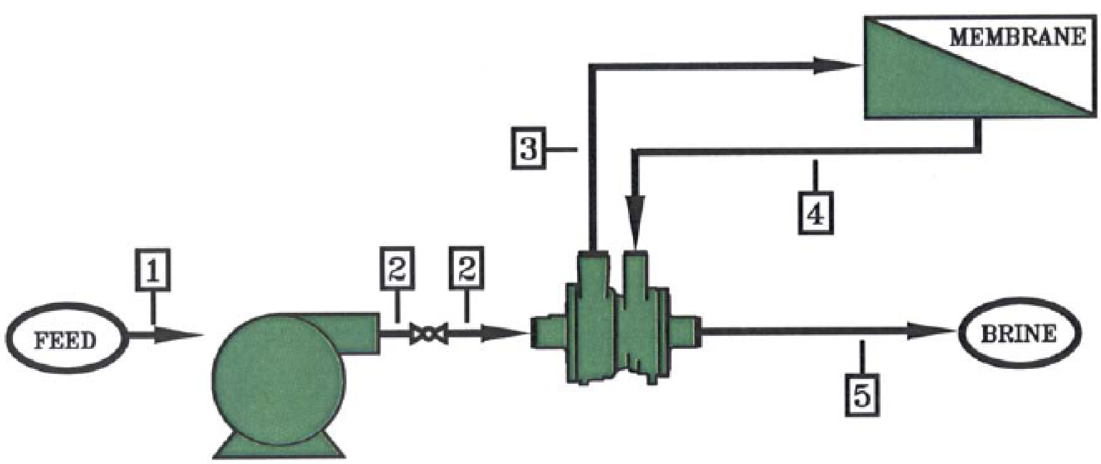


Figure 14
Flow Schematic

The Turbo Boost Pressure for this example is calculated using Equation 2 on page 8.

$$\mathbf{\Delta P = (\mathit{n}te)\ (Rr)\ (Pbr - Pe)}$$

$$\Delta P = (0.83)\ (0.60)\ (795psi) = \mathbf{396\ psi}$$

The main feed pump differential (assuming 40 psi pump suction pressure) is:

820 psi – 396 psi – 40 psi = 384 psi

Main feed pump power consumption is determined by the equation:

Q(gpm) and P(psi) (0.0005831)/n = WHP(water horsepower) or

11,500 x 384 x 0.0005831 = 2574/0.92 = 2,799 shaft horsepower

2,799 shaft horsepower x 0.746 = 2088 kW/.96 = 2175 kW wire to water power

Specific Energy Rate for the high pressure circuit = kWh/1,000gal permeate production

2175 kW/276 = 7.88 kWh/1,000gal or **2.10 kWh/m3**

6.4. Dual Turbine System

SWRO plants that employ fixed speed pumping and energy recovery equipment have to be sized for the maximum operating pressure that is projected to occur over the life of the membranes. This maximum pressure is usually the 5 year end of life condition at the time of seasonal lowest feed water temperature and/or highest salinity. So, during those months or years of operation below the maximum pressure, the pump discharge pressure is reduced by a discharge throttle valve to the level appropriate to the current membrane requirements. This throttling wastes a great quantity of hydraulic energy ahead of the membranes and reduces the pressure available to the ERD downstream of the membrane. The US and International patented Dual Turbine System is designed to eliminate feed pressure throttling associated with SWRO plant operation.

Figure 16 shows the AT 4800 in a Dual Turbine System arrangement using a Pelton impulse turbine as the secondary energy recovery unit. In this example, for simplicity and clarity, only seasonal pressure adjustments will be addressed.

Arabian Gulf SWRO experience has shown that 150 psi – 200 psi RO seasonal applied pressure variations are common. With the Dual Turbine System, the AT 4800 is used as a pressure booster working in conjunction with the high pressure pump to achieve the maximum pressure requirements of end of life conditions. Attached to a double extended shaft high pressure pump motor is a Pelton impulse turbine. Although reverse running pump turbines could be used in the DTS, the impulse turbine is the best choice for this service because of its excellent partial load capability. The secondary impulse turbine will recover brine stream energy that is not required by the AT Turbo when membrane pressure demands are less than maximum. During lower pressure operation, the AT Turbo will be operated as a variable speed (hence variable pressure) pump. The variable speed operation is achieved and controlled by the impulse turbine bypass valve that diverts brine flow from the AT Turbo to the impulse turbine. This arrangement of two turbines can offer much lower capital cost and greater efficiency than control by motor speed changes effected by more costly variable frequency drives (VFDs). VFDs have energy losses of 1.5 – 2%.

The following graph, Figure 15, based on example 2, illustrates seasonal pressure changes and the additional energy recovered by the DTS. The Pelton Impulse turbine begins operation when about 15% of its design capacity is available and ends operation when its brine flow is reduced to about 15% of design capacity. In between those two points, it will save approximately 1,208,400kW. This power savings reduces total annual energy consumption by about 9% and specific energy consumption from 3.08kWh/m3 to 2.82kWh/m3.

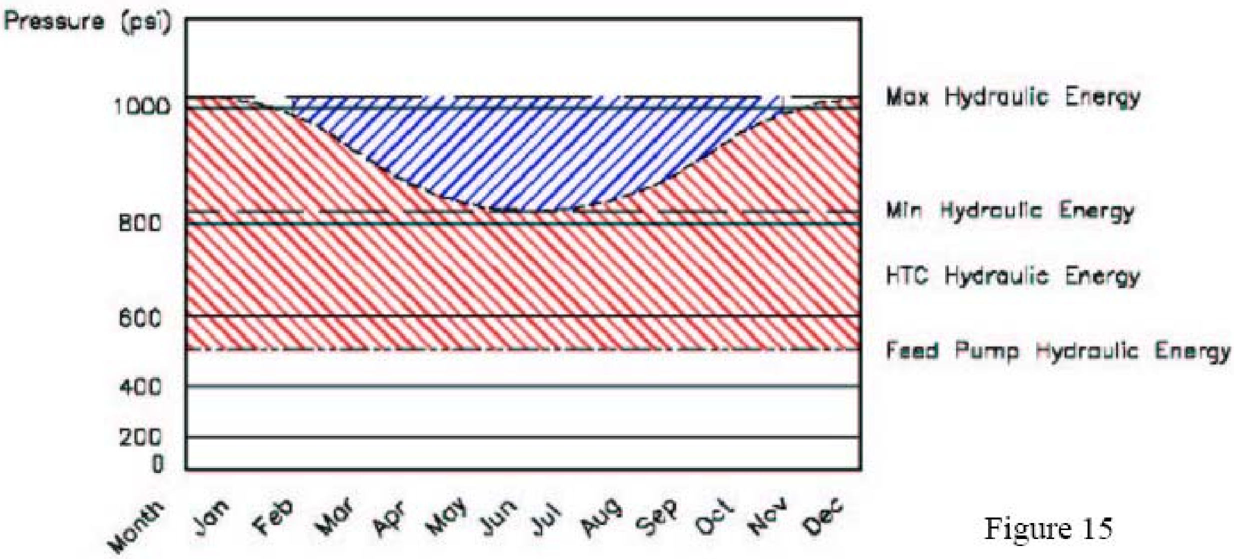


Figure 15

Energy Recovery with the Dual Turbine System – Example 2

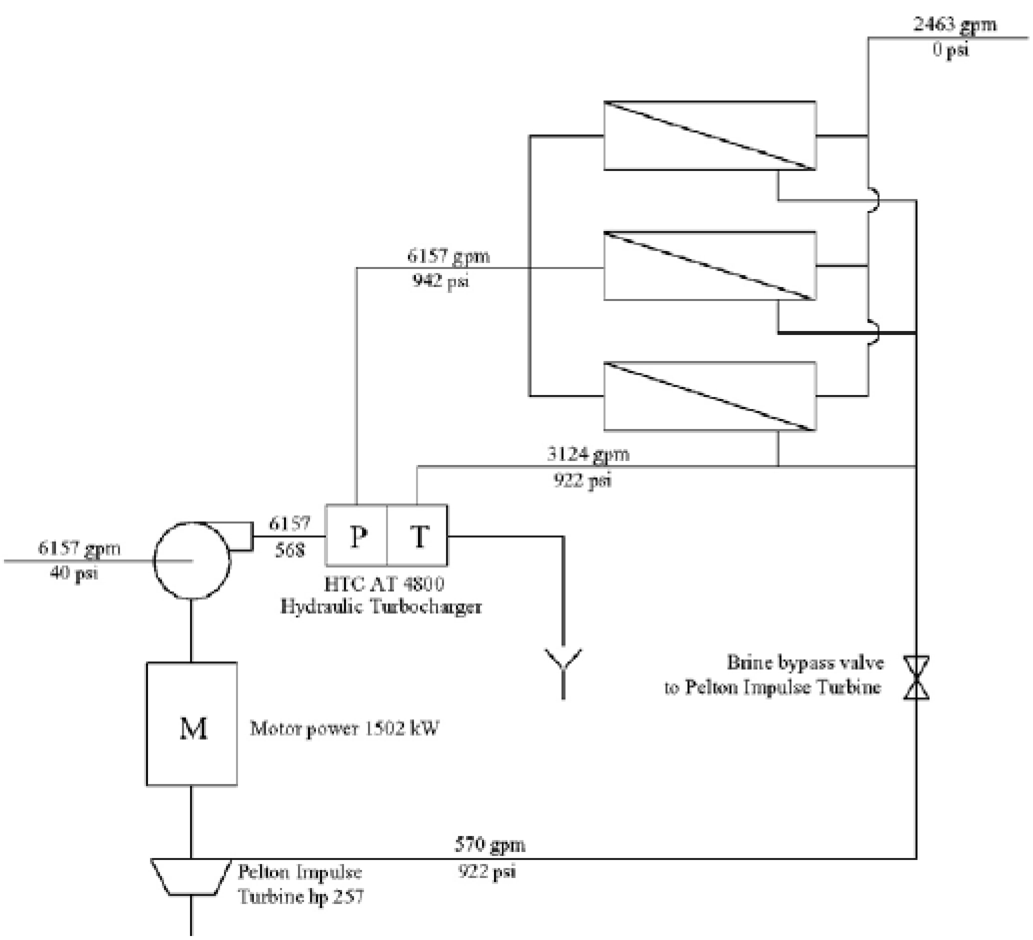


Figure 16: Dual Turbine System

VII. CONCLUSIONS

- Energy consumption is competitive with work exchangers in medium size SWRO trains and superior in larger SWRO trains.
- Capital reduction occurs as a result of:
  1. The boost pressure provided by the AT Turbo reduces the discharge pressure the main feed pump must produce, thereby, allowing for a pump with fewer stages or even single stage pumps.
  2. Motor size is reduced by 30% - 50%.
  3. Smaller transformers and motor starters or variable frequency drives (VFD) are allowed.
  4. The AT Turbo is able to discharge brine against any backpressure and a brine disposal pump/ sump/trench is not needed.
  5. The Turbo can be located anywhere, including next to the RO trains, reducing the amount of high pressure piping.
  6. The small footprint and Victaulic pipe connections ensure easy installation.
- The Dual Turbine System can eliminate all control pressure throttling losses.

VIII. REFERENCE

1. "WHAT SEAWATER ENERGY RECOVERY SYSTEM SHOULD I USE? – A MODERN COMPARATIVE STUDY" BY I. MOCH, JR. AND C. HARRIS, PRESENTED AT THE IDA WORLD CONGRESS, BAHRAIN, MARCH 2002. – Received Conference best paper award

IX. AUTHOR REFERENCES

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