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Minimizing RO energy consumption under variable conditions of operation

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Abstract

Specific energy consumption (SEC) for reverse osmosis (RO) desalination systems has usually been estimated using simplistic analyses that consider an average duty point of operation for a certain plant. A more sophisticated and comprehensive approach that accounts for the effects of variable parameters of operation on SEC was recently described, introducing the concept of the "hydraulic envelope". Variable parameters include flow rates at variable recoveries, feed temperature and salinity with their resulting pressure requirements, pressure losses caused by membrane fouling, and pressure losses caused by system controls such as feed throttle valves. This paper will explore in greater detail various energy recovery strategies under variable parameters of operation. Particular attention will be paid to a recently developed, innovative energy recovery configuration that uses a motor-driven booster pump coupled to a Pelton turbine, the so-called 'PROP', instead of a single-component high-pressure feed pump. This new energy recovery concept can not only be applied to single-stage RO plants, but also as a highly effective interstage booster for dual-stage Brine Conversion Systems (BCS). The concept has been submitted for patenting. Results of the analysis suggest that the key issue for minimizing SEC is to control the plant over the entire width of the operational range without creating throttling losses. This can only be achieved by using hydraulic equipment that allows for feed pressure adjustment at minimum energy dissipation and eliminates the need for throttling valves. It is shown that the newly developed 'PROP' concept provides minimum SEC over the entire range of the hydraulic envelope of a plant, while at the same time allowing for maximum hydraulic dynamic control efficiency of the RO unit. In addition, the 'PROP' offers a significant savings potential in terms of capital costs compared to conventional energy recovery strategies.

Keywords: Reverse osmosis; Plant control; Energy recovery; Variable feed conditions; PROP

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1. Introduction

In the reverse osmosis (RO) process a key figure of merit is the specific energy consumption (SEC), generally defined as the total energy consumed to produce a unit volume of permeate. Currently, nearly all seawater reverse osmosis (SWRO) plants do use some form of ERD in order to minimize SEC, but energy still remains the largest single operating and maintenance item for a desalting plant, exceeding in most cases 50% of the total cost [1].

The specific energy consumption of RO plants is largely dominated by two factors: the amount of transmembrane pressure difference required in order to achieve the necessary permeate flow rate at various mass transfer conditions, and the design and efficiency of the feed water pump in combination with the respective energy recovery system. For a given recovery rate, the required feed pressure is determined by the feed water properties, mostly temperature and feed salinity, which may vary significantly throughout the year due to seasonal influences. Feed water properties together with hydraulic losses from feed to brine also determine the outlet pressure of the membrane array, i.e. the inlet pressure for the energy recovery device.

For the reasons outlined above, real-life operation of an RO plant will always be determined by variations in feed water conditions, actual plant state, and actual permeate requirements. This means that the plant is operated within a certain range of each of the following operational parameters [2–4]:

- feed, brine, and permeate flow rate (governed by the desired recovery rate)
- feed and brine pressure (governed by salinity, temperature, membrane condition, and recovery rate)

Instead of a fixed duty point, there will be a duty range, which in many cases can be very wide. In contrast to this, the hydraulic components like pump and energy recovery device do have a very specific duty point in terms of flow rate and pressure, at which they will perform with optimum energetic efficiency. The further away from the respective duty point, the worse the energetic efficiency will be. The duty range of an RO plant can be described using the concept of the "hydraulic envelope" [3,4], which has been recently introduced.

A second, even more important aspect is that as the operating conditions vary, the plant hydraulics has to be adjusted accordingly over the duty range. Feed pressure and flow rate have to be increased or decreased in order to meet the requirements determined by the membrane stage for a given permeate production rate. In order to adjust the operating conditions on the feed side appropriately, a suitable control device has to be implemented. In most cases, this comprises either a throttling device or a Variable Frequency Drive (VFD). Since the physical principle of a throttling device is to dissipate energy, this limits and sometimes even counteracts the effects of the Energy Recovery Device installed to a sometimes substantial degree.

The problem faced by today's designer, specification engineer and end-user is the selection of the one plant control and energy recovery technology that is best suited for the plant being considered. Basically, this means to take into account the entire range of operational parameters during design when evaluating specific ERDs and plant control strategies. In order to do this, the first step is to define the so-called "hydraulic envelope", i.e. the duty range of a certain plant, which is strongly affected by the basic design concept and the geographical location of the system.

2. Characterization of the hydraulic envelope

The hydraulic envelope of an RO system is defined as the area bounded by the extremes of the operational parameters that an RO system may see over its lifetime of operation. The hydraulic variables feed flow rate, brine flow rate, and permeate flow rate are basically governed by the desired recovery rate and of course permeate demand. The hydraulic variables feed pressure and brine pressure are governed by feed salinity and temperature, but also by the plant hydraulics, i.e. array losses, longitudinal membrane pressure losses, throttling losses, and recovery rate [4].

If the same plant with the same membrane configuration is operated at a different recovery ratio, a new and different envelope governs for each recovery. Operating a plant at variable recovery ratios can for example be favorable in case of a strong seasonal variation of feed water salinity and/or temperature. If for example salinity is particularly low, then increasing the recovery by decreasing the required feed flow rate leads to cost savings in raw water pretreatment. Another typical reason for applying a variable recovery operating strategy may be seasonal variations in permeate demand. Fig. 1 displays the three-dimensional shape of the hydraulic envelope for a plant, taking into account the effects of variable recovery.

The hydraulic envelope displayed in Fig. 1 reflects the range of hydraulic conditions in terms of flow rates and pressures that a given plant can operate at. Each point within the envelope is a valid duty point of the plant. For the plant operator, there is a variety of reasons to move the plant from one duty point to the other, examples being a change in permeate demand, membrane cleaning requirements, overall energy consumption, and many more which are specific for each case of application. A more detailed and thorough description of the concept of the hydraulic envelope can be found in the literature [5].

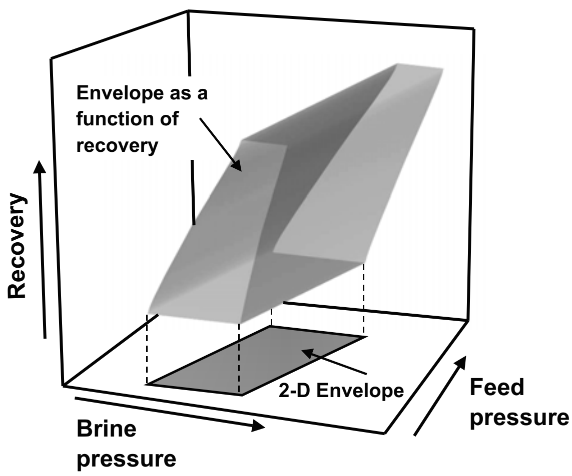

Fig. 1. Three-dimensional hydraulic envelope at variable recovery rate.

3. Energy recovery devices and operating strategies

3.1. Energy recovery devices

While the operating conditions for larger seawater RO plants clearly indicate a centrifugal type feed pump, the selection of a suitable energy recovery device (ERD) for such a plant is not so obvious. Energy recovery devices can be classified by their operating principle by which they recover the remaining energy from the brine stream of the membrane array. This operating principle can be either centrifugal or positive displacement action. Table 1 summarizes the major relevant energy recovery systems that are available on the market [1].

Out of the listed devices, reverse running pumps represent the most ineffective technology. RRPs can be found in older plants, but are no longer a consideration in the design of today's state-of-the-art RO plants. Pelton impulse turbines and turbochargers clearly are the most efficient and thus most widely used centrifugal machines

Table 1
Industrial-scale ERDs [1]

Centrifugal devices	Positive displacement devices
Reverse running pump (RRP)	Dual work exchanger (DWEER)
Pelton impulse turbine (PIT)	Pressure exchanger (PX)
Hydraulic turbocharger (HTC)	

considered by today's RO plant designers and engineers.

Even though positive displacement devices like the DWEER and the PX do hold some interesting potential, centrifugal devices are by far the most widely used technology. More than 98% of the total RO capacity installed worldwide relies on this operating principle for energy recovery, with generally excellent field records in terms of uptime, flexibility, mechanical simplicity and sturdiness. Therefore, the scope of this analysis shall be limited to the investigation of plant operation using centrifugal devices.

3.2. Plant control

In order to control a plant, i.e. to adjust the membrane inlet pressure and flow rate in accordance with the current membrane requirements [5] (which are determined mostly by feedwater salinity and temperature, see section 2), two general principles are applied: The energy dissipation method and the energy control method.

With the energy dissipation method, the energy introduced into the system through the pump motors has to cover the actual energy requirements plus the energy dissipated. In cases where the required feed pressure and/or flow rate is lower than the respective maximum at which the plant is designed to operate, excess energy has to be dissipated either through throttling in a suitable valve, or bled from the system through a bypass. Whatever amount of energy has been dissipated is lost irreversibly and cannot be recovered in the ERD. Consequently, specific energy consumption will be higher than necessary when feed water conditions get better, and vice versa. The efficiency of recovery for the remaining energy in the brine depends on the component efficiency of the ERD itself.

With the energy control method, the energy introduced into the system through the pump motors is dosed according to the requirements of the membrane array under the current conditions. This is usually accomplished by using variable frequency drives (VFDs) on the pump motors. By varying speed and torque of the pump shaft, pump outlet conditions can be adjusted over a wide range. This way, no excess energy is consumed from the power mains, and dissipation is avoided. Consequently, specific energy consumption will be accordingly lower when feed water conditions get better, and vice versa. Again, the efficiency of recovery for the remaining energy in the brine depends on the efficiency of the ERD.

It is obvious that the wider the hydraulic envelope of a plant is, the more favourable the use of a VFD is in terms of RO energy efficiency. However, there are some drawbacks that limit the applicability of a VFD. The most significant disadvantage of a VFD is investment costs, which can sometimes be even higher than the costs for the motor and the pump itself. Therefore, the question of whether a VFD will be installed or not is often determined by the cost of energy at a certain location. Another, yet small, disadvantage is that the VFD itself has an internal electrical conversion efficiency, which is typically in the range of 97%. In addition to that, the duty point of the pump will vary with impeller speed, which slightly reduces pump efficiency. However, the wider the variation in operating conditions, the more negligible this aspect becomes when compared to throttling losses.

3.3. RO plant operating strategies

3.3.1. Control by feed throttling, recovery by Pelton turbine

One of the simplest methods to operate a seawater RO unit is to run the pump at constant speed, to obtain the required membrane feed pressure through a feed throttle valve, and to recover the remaining brine energy via a Pelton impulse turbine [4]. This configuration is depicted in Fig. 2.

Inlet pressure and flow rate of the PIT will differ in accordance with pressure drops on the feed side and selected recovery ratio. As a con-

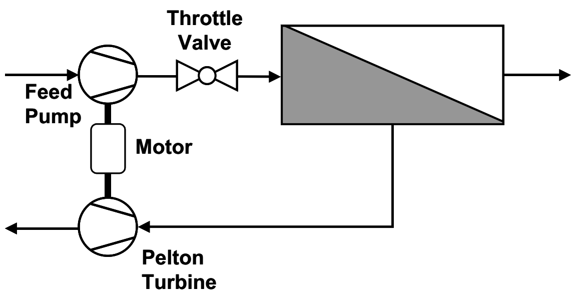

Fig. 2. Feed throttling control, Pelton turbine energy recovery.

sequence, turbine inlet pressure will vary, whereas the duty point of the pump remains constant and should be designed to match the best efficiency point (BEP) of the device.

3.3.2. Control by permeate throttling, recovery by Pelton turbine

A more sophisticated method is to run the feed pump without feed throttling and to adjust the required membrane feed pressure (or, more exactly, the required transmembrane pressure difference) through a permeate throttle valve, and to recover the remaining brine energy via a PIT [4]. Here, the control device is the permeate throttle valve, which dissipates excess energy in a pressure drop. The process diagram of this configuration is shown in Fig. 3.

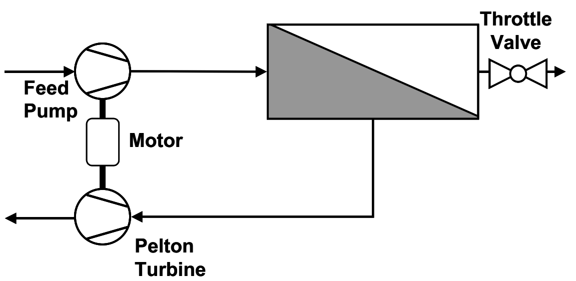

Fig. 3. Permeate throttling control, Pelton turbine energy recovery.

3.3.3. Control by VFD on feed pump, recovery by Pelton turbine

The configuration in this case is basically the same as shown in Fig. 2, except that the throttle valve is deleted and the feed pump motor is driven by a variable frequency drive (VFD) [4]. In this case, the required membrane feed conditions are adjusted and controlled via the VFD as described in section 3.2.

3.3.4. Control by throttling, recovery by hydraulic turbocharger

The hydraulic turbocharger (HTC) is an integral feed pump (single stage centrifugal) and energy recovery turbine (single stage radial inflow) [4]. The HTC returns recovered brine energy to the RO system in the form of a pressure boost in the feed stream, thereby reducing the required feed pump discharge pressure. Feed pressure control is accomplished by bypassing a portion of the high pressure brine through a small control valve, resulting in an adjustable pressure boost. This configuration is shown in Fig. 4. A variable area nozzle at the turbine side inlet of the HTC maintains the desired brine pressure on the turbine impeller.

In this way, the HTC can be adjusted to produce a pressure boost in the feed stream that equals the required membrane pressure (which can vary widely) minus the feed pressure delivered by the centrifugal feed pump (which is constant). All the time, the feed pump is operated at a

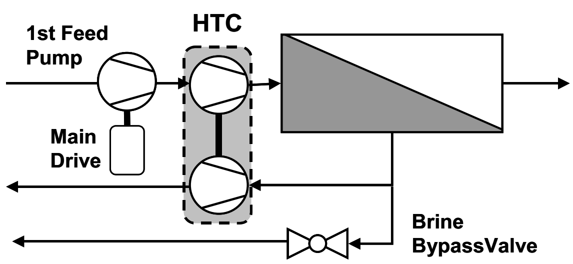

Fig. 4. HTC with brine valve control.

constant duty point, which should be the BEP. The control device is the bypass valve, which discharges excess energy, in conjunction with the variable area nozzle.

3.3.5. Control and recovery by turbocharger and VFD

As an alternative, a VFD can be used to control feed pump pressure via regulation of pump operating speed, which eliminates the need for the brine throttle valve. This turns the system displayed in Fig. 4 from energy dissipation to energy control. In this case, the full brine stream is fed through the HTC, resulting in a pressure boost on the feed side that is nearly constant and only slightly affected by a small shift around the performance curve of the HTC with varying pressure. The feed conditions for the HTC are varied through the VFD on the feed pump.

3.3.6. Control and recovery by turbocharger and helper turbine

Combining the HTC and the PIT in the same system is another way to minimize throttling [6,7]. This is possible as the units have complementary hydraulic characteristics that can fully control membrane and concentrate pressure without resorting to feed throttling or VFDs. The throttling process imposed on the bypass brine stream as depicted in Fig. 4 can be avoided by directing the bypass brine though a “helper turbine” which is connected to the high pressure feed pump shaft. In this configuration, the variable area nozzle of the PIT together with the HTC’s variable area nozzle permits splitting of the brine flow to the respective devices in accordance with the hydraulic requirements. However, the use of this concept is limited, since the PIT can only handle a limited range of flow rates. Thus only a part of the dynamic range can be covered. If the flow through the PIT is below the limit, part of the brine has to bypassed of throttled on the brine outlet of the HTC. The use of an HTC with a PIT in parallel is subject to a patent held by Pump Engineering, Inc. [6].

4. The PROP — a new control and recovery concept

4.1. Single-stage RO plants

As outlined in section 3.2, plant control through dissipation should be avoided in case of a wide hydraulic envelope, because it introduces excess energy into the system that is not actually required when the operating conditions vary. On the other hand, the capital costs that come with the installation of a VFD are exceptionally high. The basic idea behind the PROP concept (*P*elton-driven *RO P*ump) is to take the most reliable and field-proven ERD equipment and rearrange the components in such a way that dissipation is avoided while at the same time the costs for the VFD are significantly reduced. The use of the PROP concept is subject to a patent application.

Generally speaking, the total energy required to operate the RO stage can be divided into a basic constant energy consumption, determined by the minimum operating conditions, and a variable dynamic part, determined by the range between minimum and maximum operating requirements. The PROP follows this logic by splitting the high-pressure pump in two, and fitting the Pelton turbine onto the shaft of the second-stage booster pump which is driven by a separate motor using a VFD. The configuration is depicted in Fig. 5.

The first pump is operated at a constant duty point, which is the BEP of the pump. This pump thus handles the basic constant energy input required. The booster pump, with the help of the

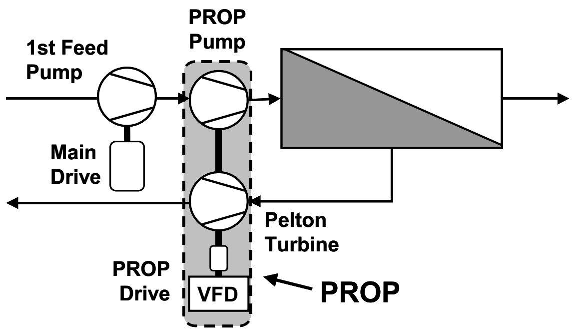

Fig. 5. Control and recovery using the PROP.

turbine and the motor, increases the feed pressure to the desired level. The booster pump thus handles the variable dynamic part of the energy input. The design of the two feed pumps in series has to be selected in accordance with the hydraulic operating requirements for a certain plant. For the minimum feed pressure and flow rate required, the energy recovered from the Pelton wheel is almost sufficient to drive the booster pump, with only a marginal additional energy input from the VFD and the motor. For the maximum feed pressure and flow rate required, the additional energy input from the VFD and the motor is at its maximum. Compared to a classic PIT/VFD configuration as described in section 3.3.3., the plant can be operated over the same dynamic range, but instead of feeding the total energy input through the VFD, the PROP only feeds the variable part through the VFD. This leads to a significant decrease in the necessary size of the VFD.

Control of the plant is accomplished through the VFD, without any throttling. The width of the hydraulic envelope determines the sizing of the second motor and its VFD. Typically, the power requirement of the second motor, and consequently the power requirement of the VFD too, is only a fraction of what would be required in a "classic" PIT/VFD single-pump configuration. Moreover, the absolute energy losses due to VFD conversion efficiency are decreased to the same amount.

4.2. Dual-stage RO plants (brine conversion systems)

Another field of application for the PROP are dual stage plants, where the concentrate of the first membrane stage is boosted to an ever higher pressure and fed into a second membrane stage. Ideally, the transmembrane pressure should remain constant along the length of the membrane channel. However, the rapidly increasing osmotic pressure due to concentration conspires against this objective. Therefore, the designer is often forced to accept undesirably high feed transmembrane pressure difference for the initial membrane elements in single-stage plants, simply to ensure that the remaining transmembrane pressure difference is sufficient for the last elements to produce an adequate amount of water. Not only is this a waste of feed energy, but also may accelerate fouling by overfluxing of the initial elements.

The brine conversion system (BCS) represents an attempt to compensate for this drawback by using a two-stage membrane configuration, with the second stage operating at a higher pressure. This configuration is applicable for plants with low to moderate salinity [8]. So far, only the hydraulic turbocharger was applicable to produce the additional boost in the feed stream for the second stage, driven by the brine from the same stage. The configuration is depicted in Fig. 6.

It is easy to envision how the application of the PROP to a BCS configuration in place of the HTC as shown in Fig. 7 adds significant benefits to this concept.

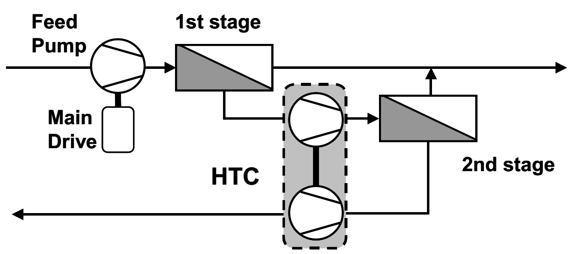

Fig. 6. BCS configuration using the HTC.

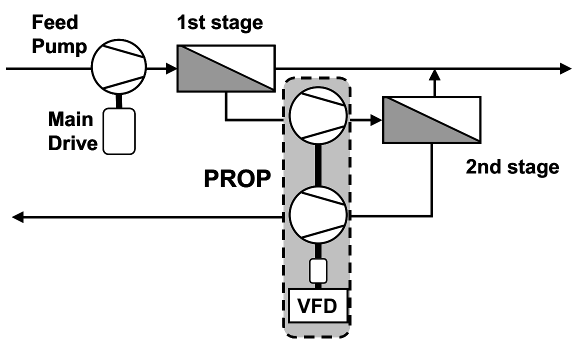

Fig. 7. BCS configuration using the PROP.

When using the HTC in a BCS, the feed pressure for the second stage is determined by the brine pressure coming from the first stage, together with the recovery rate in the second stage (which determines the brine flow rate driving the HTC). Therefore, there is no way to control the operating conditions in the second stage separately, except by bleeding part of the first-stage brine before it enters the HTC. In contrast to this, using the PROP in a BCS plant allows for separate control of the feed pressure for the second stage. This way, the second stage can be operated at its optimum without the limitations that come from the hydrodynamic conditions in the first stage.

The benefits of using the PROP in BCS plants are tremendous, since overall permeate recovery and operating duty points of the two respective membrane stages can be adjusted in a very efficient way. A detailed discussion of the savings potential and the operational benefits of a PROP in a BCS plant will be subject of a future paper.

5. Comparison of different configurations for single stage plants

5.1. Basic assumptions

In order to evaluate the different energy recovery and control configurations outlined in sections 3 and 4, they will be investigated and compared with respect to their specific energy consumption under given operating conditions. To provide a consistent comparison, a single-stage case study plant that has been investigated in a previous paper [4] is taken as an arbitrary basis. This plant is based on a case study for an RO desalination project in the Arabian Gulf producing water for irrigation purposes. In terms of salinity and temperature, this analysis can be considered representative for seawater conditions in the Arabian Gulf. Table 2 shows the relevant operational parameters for feed water conditions as determined from on-site samples; respective membrane feed pressures have been calculated from commercially available membrane design software. These conditions have been used for all the model calculations.

Table 2
Hydraulic envelope for case study plant

	Minimum	Average	Maximum
Feed salinity, g/l	45.2	45.2	44.9
Feed temperature, °C	18	26	36
Required feed pressure, bar	66.5	73.4	82.1

Table 3
Hydraulic assumptions for model calculations

Operating conditions	According to Table 2
Feed flow rate, m^3/h	950
Permeate recovery, %	40
Motor efficiency, %	97
VFD efficiency, %	97
Feed pump inlet pressure, bar	2
Feed pump type	Centrifugal
Brine discharge pressure, bar	0
Array/manifold loss, bar	4

The hydraulic assumptions for the case study plant are summarized in Table 3. Feed pump inlet pressure was assumed to be constant at 2 bar, resulting from the feed pretreatment process. Hence the pressure differential to be delivered from the feed pump can be calculated from the required feed pressure minus the inlet pressure. Axial pressure losses due to manifolds and the membrane array itself was assumed to be constant at 4 bar. Hence the pressure differential to be recovered by the ERD can be calculated from the required feed pressure minus the array losses minus the discharge pressure, which was assumed to be 0 bar. Since recovery rate is assumed to be constant at 40%, the data in Table 2 also gives the boundary values for the Hydraulic Envelope of the case study plant. Energetic efficiencies pump motors and — where applicable — the VFDs were assumed to be constant at 97%, a value that was determined from several equipment manufacturers.

This analysis only considers net energy consumption of the RO stage itself. Energy consumption attributable to the pretreatment system is constant for all cases considered in this paper (due to constant feed and permeate flows) therefore it will have no effect on the analysis.

5.2. Hydraulic hardware

5.2.1. Feed pump

Centrifugal feed pumps deliver a pressure rise that is dependent on rotor speed, flow rate and density of the pumpage. The shape of the head-capacity and head-efficiency curves are determined by the specific speed of the pump and details of the impeller, diffusion system and quality of manufacture. The feed pump used for this analysis was sized to accommodate the maximum membrane pressure requirement for the given flow. A ring section pump, supplied by Duechting Pumpen GmbH of Witten, Germany, was selected that gave a maximum energetic efficiency of 85% at the flow rate considered here. For systems using the standard Pelton Impulse Turbine (PIT) configuration, four stages are applicable. For the PROP and the HTC, the feed pump would require two stages to deliver a reduced pressure but otherwise has exactly the same efficiency vs. flow characteristic. The nominal speed was 3000 rpm for all cases. However, changing the operating speed using a VFD changes the pump performance curves in accordance with "affinity laws". The recalculation of pump performance at each possible operating speed has been included in this analysis, made practical by recently developed software.

5.2.2. Pelton wheel

The Pelton impulse turbine (PIT), a widely used ERD, typically consists of a rotor located between bearings in a housing vented to the atmosphere. A high velocity concentrate stream is directed against the buckets mounted on the periphery of the rotor to generate rotary motion on the shaft. Concentrate is discharged at atmospheric pressure. The PIT has an integral nozzle control valve that eliminates the need to bypass or throttle concentrate flow, thereby assuring that the entire hydraulic energy of the concentrate stream is available to the PIT. The performance curve of the Pelton wheel used in this analysis is given in Fig. 8. The PIT is supplied by Duechting Pumpen GmbH of Witten, Germany. The curve is based on 3000 rpm speed of operation. In that the affinity laws apply to centrifugal pumps and to PITs in the same way, the efficiency curve can be normalized to account for changes in operating speed that will occur when a (VFD) is used to control pump (hence PIT) operating speed.

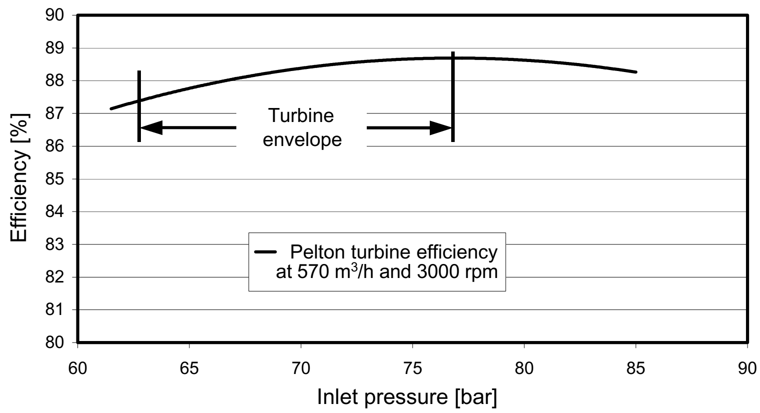

Fig. 8. Pelton turbine efficiency curve.

5.2.3. Hydraulic turbocharger

As previously described, the hydraulic turbocharger (HTC) features a single stage centrifugal feed pump and single stage radial inflow energy recovery turbine on one shaft mounted in one housing. For the present analysis, there is only one HTC on the market, manufactured by Pump Engineering, Inc., that can handle the flow rates considered here. For this size, the overall energy transfer efficiency, which includes the hydraulic losses in the feed pump section as well as in the turbine, was given to be 71%. This number applies for a reject flow ratio (i.e. brine flow rate through the HTB/feed flow rate) of 60%. Note that the reject flow ratio is not directly linked to the system recovery, since the brine flow rate through the HTB can be adjusted via a parallel valve in accordance to the required boost in the feed stream. Consequently, the transfer efficiency decreases when brine is bypassed in order to control the plant as described in section 3.

5.3. Results of analysis

The key figure to assess the energetic efficiency of a reverse osmosis configuration is the specific energy consumption (SEC), which gives the amount of energy required to produce one unit volume of permeate. A computer program was developed to deal with the complex calculations necessary to determine the efficiency of the various hydraulic components under a wide range of pressures and operating speeds. The hydraulic components selected for the analysis are commercially available products, with typical performances. The pumps used in all of the cases were optimized for the required hydraulic conditions. The performances of the ERDs were matched to the operating conditions as best as possible, but some further optimization may be possible. The analysis is believed to have accounted for all major hydraulic parameters. Other factors such as variation in motor efficiency as a function of load and changes in hydraulic machinery efficiency due to changes in fluid temperature and density were not considered. Also, no external energy consumption was considered.

5.3.1. Theoretical energy consumption

The theoretical energy consumption is the minimum amount of energy that has to be introduced in order to produce the desired amount of permeate. The theoretical specific energy consumption (TSEC) value represents the absolute minimum amount of energy for a given recovery, assuming that the efficiency of each and every component such as pump, ERD, motor, and VFD, is 100%. Under this assumption, the minimum theoretical specific energy consumption can be calculated strictly from the pressures and flow rates listed in section 5.1. For the three cases considered here, the theoretical values are listed in Table 4.

Another value of more practical interest is the minimum hydraulic specific energy consumption (HSEC). The HSEC value represents the absolute minimum amount of energy for a given recovery, assuming that the pump and the ERD are operated at their maximum efficiency in all cases, while electrical losses due to motor and VFD losses, are omitted. Again, this value can be calculated from the pressures and flow rates listed in section 5.1 and listed in Table 5. Maximum pump efficiency was taken to be 85%, maximum ERD efficiency was taken to be 89% (PIT).

A comparison between the values given in Tables 4 and 5 shows that even when high-efficiency components are used and electrical losses are neglected, the HSEC is more than 50% higher than the TSEC. The reason for this effect is of course that losses due to non-ideal efficiencies are amplified by the recovery rate factor.

5.3.2. Actual energy consumption

In contrast to the theoretical considerations and assumptions outlined above, real life plants are less energy efficient due to electrical conversion losses, but also due to dissipation in the control devices used (see section 3). The amount of

Table 4
Theoretical specific energy consumption (TSEC) for case study plant

	Minimum	Average	Maximum
Theoretical specific energy consumption, kWh/m^3	1.96	2.07	2.31

Table 5
Minimum hydraulic specific energy consumption (HSEC) for case study plant

	Minimum	Average	Maximum
Minimum hydraulic SEC, kWh/m^3	3.03	3.26	3.64

Table 6
Specific energy consumption (HSEC) for case study plant (kWh/m^3)

	Minimum	Average	Maximum
TSEC	1.96	2.1	2.3
HSEC	3.03	3.26	3.64
PIT, feed throttling	4.35	4.13	3.74
PIT, permeate throttling	3.74	3.74	3.74
PIT/VFD	3.22	3.49	3.88
PROP	3.14	3.41	3.82
HTC	3.96	3.96	3.96
HTC/PIT	3.38	3.6	3.96
HTC/VFD	3.33	3.65	4.07

additional energy requirement depends on the specific control and recovery strategy. For the configurations described in sections 3 and 4, the SEC was calculated. Results are given in Table 6.

Fig. 9 shows a comparison between the respective configurations at minimum feed pressure requirements. Since in this case the difference to the highest feed pressure requirements is maximum, systems using a throttling device or a bypass for control have by far the highest SEC, because dissipation energy losses have to be introduced into the system via the motors. For the HTC with the PIT helper turbine, the brine energy can be recovered in the turbine. Yet systems using a VFD for control prove to be superior. SEC for the PROP and the Pelton turbine with a VFD are almost the same, with a slight advantage for the PROP. This is because here the absolute conversion losses are lower due to the smaller size of the VFD, and because the first feed pump of the PROP configuration is operated at its best efficiency point (BEP), while the feed pump for the PIT/VFD configuration and the PROP pump are operated slightly off the BEP in order to adjust the required feed conditions.

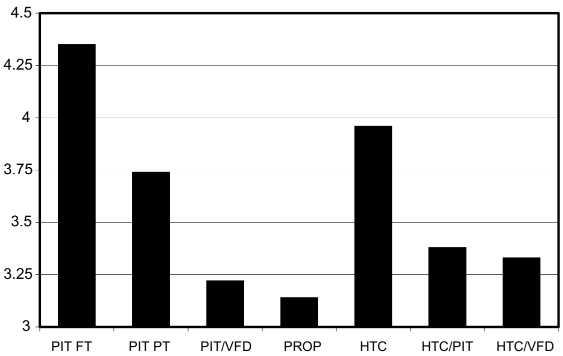

Fig. 9. Specific energy consumption for case study plant — minimum conditions.

Fig. 10 shows the same comparison for the case of average feed pressure requirements. Even though feed pressure requirements are lower than in Fig. 9, SEC for the PIT with feed throttling is

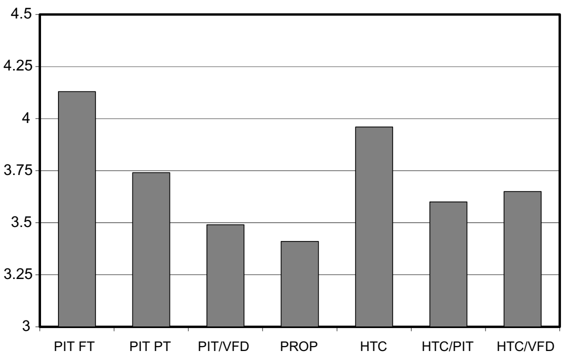

Fig. 10. Specific energy consumption for case study plant — average conditions.

lower. This is due to the fact that since less energy is lost in the throttle valve, more energy can be recovered in the turbine. For the HTC and the PIT with permeate throttling, SEC remains the same, since the feed pump is driven at constant energy input, part of which is bled from the system via the bypass or the permeate throttle valve. For the HTC with the PIT Helper Turbine, less brine energy than in the minimum case can be recovered in the turbine, resulting in an increased SEC. Systems using a VFD also show this logical behaviour of increasing SEC with increasing feed pressure requirement. Again, the PROP shows the lowest SEC due to the same reasons outlined in Fig. 10.

Fig. 11 shows the comparison for maximum feed pressure requirement. Again, the SEC for HTC system and the PIT with permeate throttling remains constant for the reasons explained above. SEC for the PIT with feed throttling becomes minimum and equals the SEC for the PIT with permeate throttling, since no throttling at all occurs. SEC for the systems using a VFD for control becomes maximum when feed pressure is maximum. Interestingly, the PIT using a VFD has a higher SEC than the PIT systems without; the difference reflects the conversion losses in the VFD. Again, the PROP is slightly lower because of its smaller VFD. The SEC for the HTC with and without the helper PIT are equal, since no

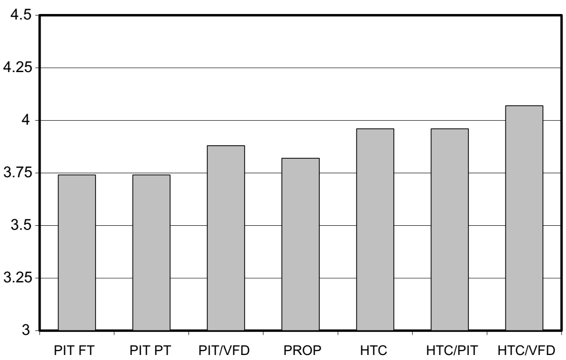

Fig. 11. Specific energy consumption for case study plant — maximum conditions.

bleed flow goes through the turbine. This figure reflects the net transfer efficiency of the HTC in case no throttling and no bypassing occurs.

5.3. Summary of analysis

The analysis outlined above clearly indicates a few guidelines for the selection of a suitable control and recovery strategy for a given RO system.

When the system is operated at a constant duty point, with no seasonal variations in feed conditions and the resulting feed pressure requirements, the best ERD is a Pelton turbine, since it delivers maximum recovery efficiency.

When feed pressure requirements are variable, i.e. when the plant has a considerable hydraulic envelope, throttling is about the worst one can do in order to control the plant and adjust it to the varying feed conditions, because inevitable pressure losses drive the energy consumption the more upward the more favourable the conditions get. Slightly better is a PIT with control through permeate throttling, yet this option is often considered critical due to the potential of membrane damage from permeate backpressure. When even a small VFD is not applicable due to local restrictions, the use of the HTC coupled to a helper PIT will provide the minimum SEC at a decent control range. Yet when the range of operating conditions is changing throughout the lifetime of any plant, a VFD is the best selection for control. Based upon the advantages described in section 4, the PROP will provide the required control dynamics at minimum capital and energy costs when compared to a classic PIT/VFD configuration.

6. Conclusions

Seasonal variations in feedwater conditions such as temperature and salinity impose changes on the conditions under which a plant is operated, namely the feed pressure required to produce a certain amount of permeate at a given recovery rate. Hydraulic hardware built into an RO plant has to be able to deal with all conditions that are allowed for a specific plant. Adapting the plant to variable parameters of operation requires a control strategy. This can be done either by using a VFD or by throttling. The wider the hydraulic envelope of a certain plant is, the more important the adaptability of a certain ERD becomes. The wider the hydraulic envelope of a specific plant in terms of allowable pressures and recoveries is, the more savings potential exists in terms of SEC if adaptable equipment is used.

A variety of plant control and energy recovery strategies was described, investigated, and evaluated. The selection of a suitable ERD and control strategy is a complex process for each specific plant. There is no such thing as the best ERD. Any given ERD can provide particularly low SECs at a certain duty point, but may look significantly less favorable under different duty conditions. Therefore, the hydraulic and energetic behavior of commercially available devices has to be investigated in light of the specific operational conditions of each plant.

Results of the analysis suggest that the when operational parameters of a plant vary, the key issue for minimizing SEC is to control the plant over the entire width of the operational range without creating throttling losses. This can only be achieved by using hydraulic equipment that allows for feed pressure adjustment at minimum energy dissipation and eliminates the need for throttling valves. Throttling or bleeding dissipates energy and increases specific energy consumption. This effect was proven to be the more significant the more favourable mass transfer conditions become. The benefits of the application of frequency converters driving the pump was investigated as an alternative solution. Systems using a VFD for control and a Pelton turbine for energy recovery proved to be clearly superior.

It is shown that the newly developed PROP concept provides minimum SEC over the entire range of the hydraulic envelope of a plant, while at the same time allowing for maximum hydraulic dynamic control efficiency of the RO unit. In addition, the PROP offers a significant savings potential in terms of capital costs compared to conventional energy recovery strategies.

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