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The design aspects of rotary work exchanger for SWRO

Osamah M. Al-Hawaj
Mechanical Engineering Department, College of Engineering, Kuwait University, P.O. Box 5969, Safat 13060, Kuwait
Fax (965) 484-7131; email: osamahalhawaj@yahoo.com

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ABSTRACT

Seawater reverse osmosis plants, SWRO, are currently becoming increasingly more important compared to the less efficient MSF plants. The overall efficiency of SWRO plant is strongly dependant on the type of energy recovery device, ERD, used. Work or pressure exchangers, PE, are among the well proven devices for efficient energy recovery. The rotary work exchanger, RWE, technology is an improved pressure exchange concept characterized by its simple construction, high-pressure transfer efficiency and design flexibility. This paper discusses the operation and some of the design aspects of the RWE. Also, the paper presents a mathematical model for predicting the specific energy consumption and pumping efficiency of SWRO plant employing the RWE. Furthermore, the study performs a parametric analysis illustrating the significant effect of efficiency parameters of the RWE, membrane, and high-pressure pump on the performance of SWRO plant.

Keywords: SWRO; Pressure exchanger; Work exchanger; Rotary work exchanger

1. Introduction

One major goal of the people working in the water industry is to produce potable water with an acceptable quality at a minimum cost. The desalination process for production of fresh water from seawater is known to be an expensive process because of its high energy demand. Since the advent of reverse osmosis (RO) in 1970s, tremendous effort was undertaken to find a way to reduce the associated operating costs. Among the recent advances to reduce the cost of desalinated water are the application of energy recovery devices (ERDs) and the improvement in RO membrane. The cost of energy in SWRO process account for 30–50% of the total operating cost of water and can reach 75% of the operating cost, depending on the cost of electricity [1], whereby the high-pressure feed pump accounts for at least 35% of the operating costs. Therefore, reducing the energy cost by reducing the energy wastage in the high energy brine is of vital importance.

Francis Turbine (FT) or the reverse running pump is one of the earliest forms of ERD used in SWRO plants. The FT is a reaction machine whereby part of the work done by the fluid on the rotor is due to reaction from pressure drop, and part due to change in kinetic energy, which represents an impulse function. The power transmitted to the main shaft is used to assist the main electric-motor in driving the high-pressure pump. Although, the FT scheme is characterized by its simplicity and ease of operation it suffers from a relatively low maximum efficiency (about 75%), narrow operable range at maximum efficiency, higher energy losses with no power generation when operating below 40% of the design condition [2]. Examples of plants that employ FT include SWRO plants in Saudi Arabia at Al-Jubail and Yanbu [3,4].

Another ERD used in the SWRO plants is the Pelton impulse turbine, PIT. Unlike FT, the PIT is a pure impulse turbine in which the pressure energy of the brine is converted into kinetic energy in the form of a jet issued from a nozzle impinging on a succession of curved buckets fixed to the periphery of a rotating wheel. The power transmitted to the wheel shaft is used to assist the main electric-motor in driving the high-pressure pump. The PIT system is characterized by an average maximum efficiency about 85%, flatter efficiency curve, simplicity and ease of operation and control using an adjustable input nozzle. The PIT is widely used in the desalination industry and proved to be more efficient than reverse running turbine. Examples of SWRO plants employing PIT include the Canary Islands [5]. The PIT suffers from a decline in efficiency similar to FT but at lesser extent when operating at low recovery whereby energy recovery starts at about 20% of the design condition [6].

The hydraulic Turbo Charger, HTC, is another ERD used in SWRO. It is an integral turbine-driven-centrifugal pump used in combination with electric-motor-driven HP feed pump. The system is equivalent to two pumps in series, the first pump is an electric-motor-driven pump and the second is a hydraulically driven pump, whereby the reject brine used as the driving fluid. The turbine portion of the HTC is a single stage radial inflow type hydraulic turbine and the pump portion is a single stage centrifugal with its impeller mounted on the turbine shaft. The principal of operation is basically similar to FT and PIT whereby the pressure energy of brine is converted to mechanical shaft power in the turbine section and back to hydraulic energy in the pump section resulting in a feed pressure increase. The HTC is characterized by being dynamically balanced as a unit, ease of control of membrane feed pressure using a by-pass around the HTC, and a claimed efficiency up to 70% depending on the capacity. HTC units are successfully operated in the Caribbean Island and Mas Palomas, Gran Canaria (Spain) with capacities ranging from 14,800 to 20,400 m^3/d [7].

Pressure exchanger (PX), PE, is a positive displacement ERD which can be classified as either stationary or rotary. Desalco's Work Exchanger Energy Recovery (DWEER) is an example of the stationary type, and the Energy Recovery Inc.'s PX is an example of the rotary type. The former uses a pair of cylinder fitted with floating pistons and a set of valves. The rotary PX uses a cylindrical rotor with longitudinal ducts parallel to its rotational axis. The rotor spins inside a sleeve between two end covers. Pressure energy is transferred directly from the high-pressure stream to the low-pressure stream in the ducts of the rotor. Some fluid that remains in the ducts serves as a barrier that inhibits mixing between the two streams. Both systems claim to have flat efficiency curves with efficiency ranging 91–96% [8,9]. The energy saving of PX is the result of load reduction on the HP pump. Considering a seawater RO system with a recovery rate of 40%, the ERD supplies 60% of the membrane feed flow, and negligible energy consumed by the booster accounting for less than 3% of total consumed energy. Therefore, nearly 60% of the membrane feed flow is pressurized with almost no energy input. In the PX device, pressure energy exchange is made through direct fluid contact between reject brine and an equal feed portion, while in the DWEER device pressure exchange is made across floating pistons. Higher capacities in both PX and DWEER systems are obtained by using multiple units in parallel. PXs are used at Lanzarote, Spain [10] and in Canary Island, Spain [11] both at capacity of 5000 m^3/d. DWEER upgraded an ERD system in Grand Cayman Spain [12] resulting in decrease in SEC from 3.0 to 2.22 kWh/m^3. Both PX and DWEER systems claim to have specific energy consumption (SEC) ranging from 2.0 to 2.4 kWh/m^3.

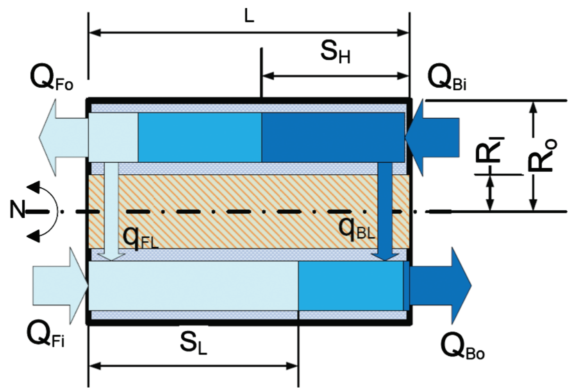

Fig. 1. Schematic flow diagram of RWE.

The rotary work exchanger, RWE, [13,14], is an alternative ERD system that operates on the positive displacement principle and uses a spinning rotor partially driven by the impulsive action of small external jet. This paper discusses the characteristics of the RWE device and presents a mathematical model describing its performance in SWRO plant.

2. Rotary work exchanger (RWE)

The RWE device is a positive displacement ERD which operates on the principle of direct pressure exchange between two differently pressurized fluids. A sectional view of the RWE device is schematically shown in Fig. 1 as one component of SWRO system. The RWE device comprises an annular rotor with a multiplicity of predominantly axial channels making openings with the inner surface of the rotor at both ends. The rotor is hydro-dynamically spinning around a central stationary core shaft and communicating through its end openings with a pair of end headers connected to respective manifolds formed into the stationary core. As the rotor makes a single revolution, each channel undergoes a forward high-pressure stroke, whereby high-pressure brine pushes the pressurized feed followed by a reverse low-pressure stroke, whereby the low-pressure seawater feed pushes the depressurized brine. The reciprocating buffer zone formed in the channel act as a barrier between brine and feed such that the pressurized feed always maintains the same concentration as the inlet seawater feed. The rotation of the rotor is effected by a combination of the impulsive and reaction effect of the flow on rotor channel openings. An alternative hydrodynamic means to effect rotation is through the use of external impinging jet on a set of buckets externally fixed to the spinning rotor. The external means of rotation is not an essential; however it can be used advantageously to control the rotational speed for best performance. Hydrodynamic lubrication is made naturally due to leakage through the small clearance formed between the inner wall of spinning rotor and the central stationary core, unlike PX system whereby hydrodynamic lubrication is employed at its external rotor surface.

3. Mass balance of RWE

The RWE device in part resembles a rotary displacement pump comprising a cylindrical block with multi cylinders, each cylinder fitted with a double head reciprocating piston, whereby the motion of the double acting pistons is controlled mechanically. In the RWE case, the pistons comprises fluid buffer zones and the reciprocating motion is controlled by the alternating end pressure differential as the rotor rotates. The motion of the buffer region for a single cylinder or channel is approximately sinusoidal in nature with flat middle portion. In this analysis, only average of all channel flow is considered such that a steady state flow condition prevails at inlet and outlet ports. The RWE device is modeled as comprising two counter current streams executing two strokes. As shown in Fig. 1, a high-pressure forward stroke of the high-pressure brine displaces and pressurizes the feed filled channels during the first half of rotor revolution, and followed by a low-pressure reverse stroke with fresh feed that displaces the depressurized brine during the second half of rotor revolution. Because of the high-pressure differential across the high and low-pressure channels, leakages occur at both ends of the channels, the brine side and the feed side, together forming a quantity, q_L, such that stream mass balances can be written as:

$$\begin{aligned} Q_{\mathrm{Bi}} &= Q_{\mathrm{Fo}} + q_{\mathrm{L}} \\ Q_{\mathrm{Fi}} &= Q_{\mathrm{Bo}} - q_{\mathrm{L}} \end{aligned} \tag{1}$$

where

$$q_{\mathrm{L}} = q_{\mathrm{BL}} + q_{\mathrm{FL}} \tag{2}$$

and the overall mass and salt balances become

$$Q_{\mathrm{Bi}} + Q_{\mathrm{Fi}} = Q_{\mathrm{Fo}} + Q_{Bo} \tag{3}$$

$$X_{\mathrm{Bi}}Q_{\mathrm{Bi}} + X_{\mathrm{Fi}}Q_{\mathrm{Fi}} = X_{\mathrm{Fo}}Q_{\mathrm{Fo}} + X_{\mathrm{Bo}}Q_{\mathrm{Bo}} \tag{4}$$

Mixing takes place within rotor channel at the interface between the fresh seawater feed and membrane reject brine as a result of turbulent eddy mixing which results in a mixing buffer region. This buffer region size is dependent on rotor speed and volumetric capacity, whereby smaller buffer zones associate with higher rotor speed and smaller capacities and vice versa. In order to prevent the buffer zone from flowing through the feed outlet, the low-pressure stroke must be greater than the high-pressure stroke (i.e. $S_{\mathrm{L}} > S_{\mathrm{H}}$), such that the RWE operate in a mode that results in flushing of the buffer zone.

Defining a volumetric efficiency η_v as the ratio of the pressurized feed outflow to the brine inflow and defining a dilution factor, ψ_d, as the ratio of the brine outflow to brine inflow, such that

$$\eta_v = \frac{Q_{\mathrm{Fo}}}{Q_{\mathrm{Bi}}} = 1 - \frac{q_{\mathrm{L}}}{Q_{\mathrm{Bi}}} \tag{5}$$

$$\psi_d = \frac{Q_{\mathrm{Bo}}}{Q_{\mathrm{Bi}}} = \frac{Q_{\mathrm{Fi}}}{Q_{\mathrm{Bi}}} + \frac{q_{\mathrm{L}}}{Q_{\mathrm{Bi}}} \tag{6}$$

Hence,

$$\frac{Q_{\mathrm{Fi}}}{Q_{\mathrm{Bi}}} = \eta_v + \psi_d - 1 \tag{7}$$

Considering an RO plant operating with recovery ratio R and feed to membrane, Q_{MF}, Eq. (7) can be rewritten to give an expression for the dilution factor as

$$\psi_d = 1 - \eta_v + \left(\frac{1}{1-R}\right)\left(\frac{Q_{\mathrm{Fi}}}{Q_{\mathrm{MF}}}\right) \tag{8}$$

An expression for the dilution factor can then be written as

$$\psi_d = \frac{X_{\mathrm{Bi}} - X_{\mathrm{Fi}}}{X_{\mathrm{Bo}} - X_{\mathrm{Fi}}} - \eta_v\left(\frac{X_{\mathrm{Fo}} - X_{\mathrm{Fi}}}{X_{\mathrm{Bo}} - X_{\mathrm{Fi}}}\right) \tag{9}$$

According to the above equations, the following remarks can be made:

1) When there is no leakage occurring at both ends of the RWE, according to Eq. (5) and Eq. (6), the volumetric efficiency is equal to unity, and the dilution factor is equal to the volumetric rate ratio of inlet feed to inlet brine. In this case three modes of operation are possible according to Eq. (9).

   (a) Dilution mode (i.e. $\psi_d > 1$) whereby the brine outflow is diluted by the excess feed and the outflow brine concentration is decreased while feed concentration remain unchanged (i.e. $X_{Bo} < X_{Bi}$, $X_{Fi} = X_{Fo}$).
   (b) Contamination mode (i.e. $\psi_d < 1$) whereby the seawater feed is contaminated with excess brine and the feed outflow concentration is increased while the brine concentration remain unchanged (i.e. $X_{Fo} > X_{Fi}$, $X_{Bi} = X_{Bo}$).
   (c) Zero mixing mode (i.e. $\psi_d = 1$) is a border line case whereby the concentration of inflow and outflow streams of both brine and feed remain unchanged (i.e. $X_{Bi} = X_{Bo}$, $X_{Fi} = X_{Fo}$)

2) According to Eq. (8), the dilution factor can be increased by either increasing the fraction of feed diverted to the RWE or by operating at higher recovery ratios.
3) Equation (9) can be used to calculate the dilution factor from measurements of stream concentrations and volumetric efficiency, or alternatively, it can be used to calculate the volumetric efficiency from measurements of concentration and dilution factor.
4) According to Eqs. (8) and (9), the dilution factor increases when operating at lower volumetric efficiencies. In other words, the effect of leakages taking place at both ends of the rotor in the RWE reduces the chances of feed contamination. Also, the minimum value of dilution factor occurs at a maximum volumetric efficiency of unity.

In practice, leakage cannot be totally eliminated but can be reduced through improved design. Although leakage reduces the overall efficiency of RWE device, small amount of leakage is needed to provide means for hydrodynamic lubrication within the clearance between the central stationary shaft and the inner wall of the rotor. Also, the small leakage allows the RWE device to operate in the safe dilution mode, whereby the dilution factor is greater than unity thereby pushing the fluctuating buffer zone further toward the outlet brine side.

4. Energy balance of RWE

An energy balance for the RWE making device can be written as

$$Q_{Bi}H_{Bi} + Q_{Fi}H_{Fi} = Q_{Fo}H_{Fo} + Q_{Bo}H_{Bo} + \Delta H_{Loss} \tag{10}$$

The energy losses in the RWE comprise fluid friction losses associated with inflow and outflow through manifolds and rotor channels, and leakage losses at both ends. Assuming equal inlet head losses and equal outlet head losses, and rotor losses to include frictional channel losses and energy required to rotate the rotor, an approximate expression for the losses can be written employing definitions given in Eqs. (5–7)

$$\Delta H_{Loss} = Q_{Bi}[(\eta_v + \psi_d)(h_{mi} + h_{mo}) + h_r] \tag{11}$$

Normalizing by the inlet brine and using the definition in Eq. (11) such that Eq. (10) can be rewritten as

$$\begin{aligned} H_{Bi} + (\eta_v + \psi_d - 1)H_{Fi} &= \eta_v H_{Fo} + \psi_d H_{Bo} \\ &\quad + (\eta_v + \psi_d)(h_{mi} + h_{mo}) + h_r \end{aligned} \tag{12}$$

A first expression for the efficiency of RWE is defined as the ratio of the hydraulic energy in the feed and brine outflows to the hydraulic energy in the brine and feed inflow, hence

$$\eta_{RWE1} = \frac{\eta_v H_{Fo} + \psi_d H_{Bo}}{H_{Bi} + (\eta_v + \psi_d - 1)H_{Fi}} \tag{13}$$

Still a second expression for the efficiency can be defined as the ratio of the hydraulic energy gain in the feed stream to the hydraulic energy loss in the brine stream, hence

$$\eta_{RWE2} = \frac{\eta_v H_{Fo} - (\eta_v + \psi_d - 1)H_{Fi}}{H_{Bi} - \psi_d H_{Bo}} \tag{14}$$

Both efficiency definitions illustrate the importance of improving the design of both inlet–outlet manifolds and inlet–outlet rotor transition passages for lower total friction and thereby improved performance from the RWE. For a dilution factor of unity, Eq. (14) reduces to the pressure energy gain in the feed stream to the pressure energy loss in the brine stream, such that

$$\eta_{RWE3} = \eta_v \frac{\Delta H_F}{\Delta H_B} \tag{15}$$

For a RWE device operating with a nearly 100% volumetric efficiency, the efficiencies as expressed by Eqs. (13) and (14) reduces to the ratio of pressure

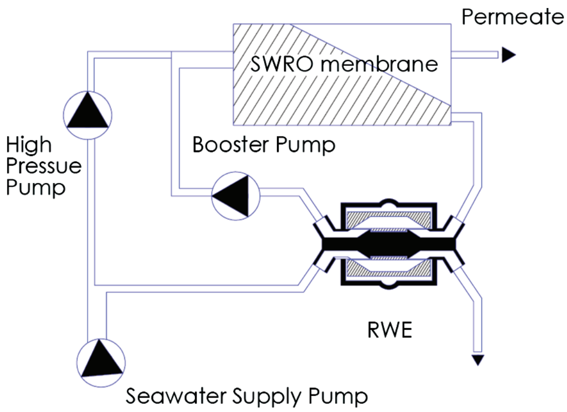

Fig. 2. Typical configurations of SWRO with RWE.

energy in the feed outflow to the pressure energy in the brine inflow, hence

$$\eta_{\mathrm{RWE}} = \frac{P_{\mathrm{Fo}}}{P_{\mathrm{Bi}}} \tag{16}$$

Defining a membrane efficiency as the ratio of the membrane reject brine pressure to the membrane feed pressure such that

$$\eta_m = \frac{P_{\mathrm{Bi}}}{P_{\mathrm{mF}}} \tag{17}$$

Considering an RO plant schematically shown in Fig. 2, which comprises a high-pressure pump with an overall efficiency, η_{HP}, and a booster pump efficiency, η_{BP}, and using a PX with an efficiency, η_{RWE}, an approximate expression for the actual specific energy consumption, SEC, can be written as

$$\mathrm{SEC} = k_1\left[\frac{1}{\eta_{\mathrm{HP}}} + \frac{1}{\eta_{\mathrm{BP}}}\left(\frac{1}{R} - 1\right)(1 - \eta_m\eta_{\mathrm{RWE}}) + k_2R^2\right] \tag{18}$$

Here the last term in the brackets in Eq. (18) accounts for the Darcy frictional effect through RO membrane, and the first constant, k_1, relate to the specific energy consumption by an ideal HP pump operating on permeate flow. The constants k_1 and k_2 are chosen approximately to correlate with results published in literature for PX class, and taken to be equal to 1.45 and 1.5, respectively. An expression for the minimum specific energy consumption can be obtained considering and an ideal RO plant having a specified membrane and maximum HP pump efficiencies ($\eta_{p\,\max} = 90\%$), and employing ERD with 100% efficiency. Hence, Eq. (18) reduces to

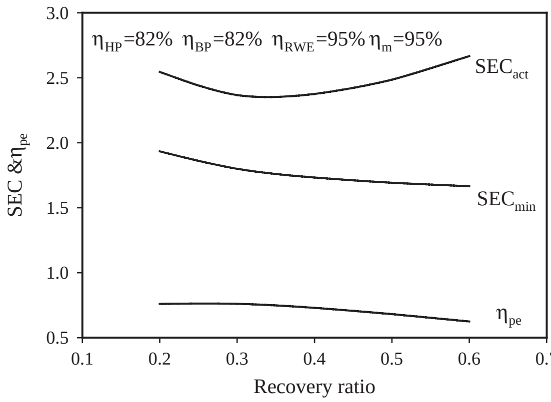

Fig. 3. Specific energy consumption and pumping efficiency for a typical SWRO plant employing RWE.

$$\mathrm{SEC_{min}} = \frac{k_1}{\eta_{p\,\max}}\left[1 + \left(\frac{1}{R} - 1\right)(1 - \eta_m)\right] \tag{19}$$

A formula for the pumping efficiency can then be expressed as the ratio of the minimum to actual specific energy consumption such that

$$\eta_{\mathrm{pe}} = \frac{\mathrm{SEC_{min}}}{\mathrm{SEC}} \tag{20}$$

Fig. 3 presents SEC for a typical SWRO with RWE as an ERD, the results are consistent with results obtained by Stover [15] and Bross and Kochanowski [16] for PX. Also, the pumping efficiency is shown to be almost flat with respect to variation in recovery ratio. Figs. 4–7 illustrate the effect of varying HP pump, booster pump, membrane, and RWE device efficiencies on the specific energy consumption. As expected, any improvement in any of the mentioned efficiencies results in reduced SEC. The improvement in SEC from Booster pump is the least significant, while the

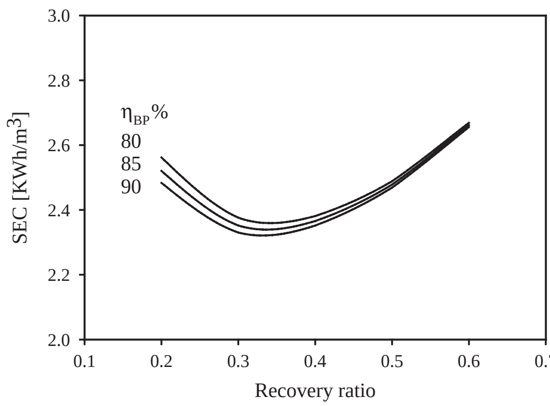

Fig. 4. Specific energy consumption, variation of booster pump efficiency.

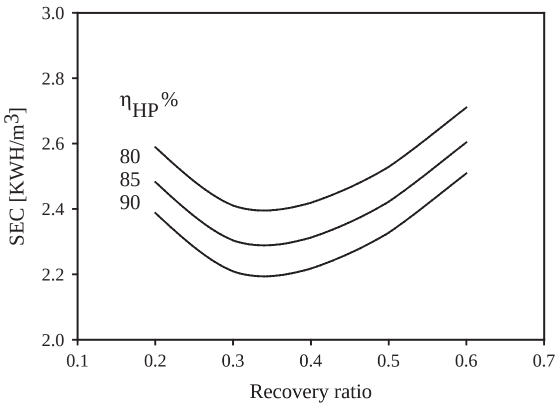

Fig. 5. Specific energy consumption, variation of HP pump efficiency.

improvement in SEC resulting from membrane and RWE are more significant when operating at lower recovery ratios.

5. Capacity of RWE

The overall dimensions of the RWE device are important for proper operation of the device as ERD. A first concern is that the length of the rotor channel must be sufficiently greater than the high-pressure stroke length with a safety margin to prevent brine from mixing with the pressurized feed. On the other hand, using very long rotor increases hydrodynamic losses and reduces RWE efficiency. This constraint can be met by requiring the rotor channel length to be greater than the anticipated high-pressure stroke length by a factor, δ, with appropriate range between 1.5 and 2.5 such that

$$L = \delta S_H \tag{21}$$

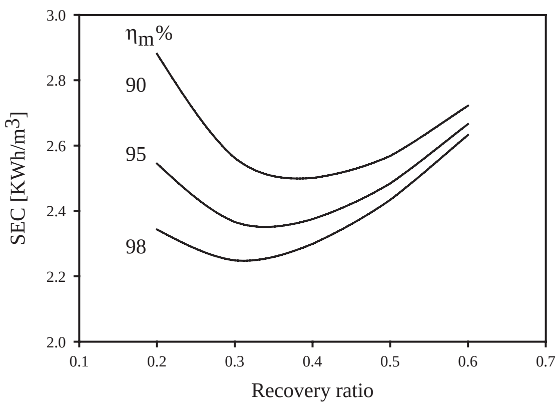

Fig. 6. Specific energy consumption, variation of membrane efficiency.

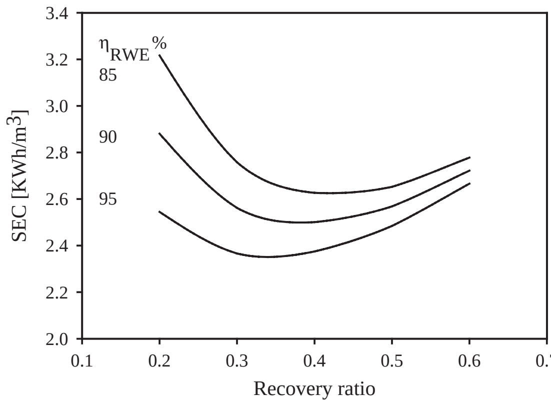

Fig. 7. Specific energy consumption, variation of RWE efficiency.

The high-pressure stroke length can be estimated for a prescribed brine inflow rate, Q_{Bi}, flowing through an effective channel area, A_{ch}, in a rotor rotating with a constant speed, N, such that

$$S_H = \frac{Q_{\mathrm{Bi}}}{A_{\mathrm{ch}} N} \tag{22}$$

The stationary cylindrical core of the RWE device includes inlet and outlet headers in fluid communication with rotor channels. To reduce inlet frictional losses resulting from average liquid velocity variation, it is appropriate to consider an inlet header cross-sectional slightly smaller than the average rotor channel area by a factor, γ, ranging between 0.5 and 1.

$$A_h = \gamma A_{\mathrm{ch}} \tag{23}$$

Neglecting channel blade thickness and assuming circular geometry, Eq. (23) gives a relation between the outer and inner radii of the rotor, such that

$$R_{\mathrm{o}} = \sqrt{\frac{\gamma + 1}{\gamma}} R_I \tag{24}$$

Also, the high-pressure stroke length, S_H, must be greater than the sum of the inlet length plus the required entrance length for the flow to reach a fully turbulent channel flow. This length is expressed as a multiple factor, β, times the inner rotor radius, with a typical range between 4 and 6.

$$S_H = \beta R_I \tag{25}$$

An expression relating the brine inflow to the inner radius can then be expressed as

$$Q_{Bi} = 0.5\pi\left(\frac{\beta}{\gamma}\right)R_I^3 N \tag{26}$$

6. Numerical example

Consider an RO plant with capacity, $Q_p = 50$ m^{3}h, 40% recovery ratio, $\eta_{HP} = 80\%$, $\eta_{BP} = 85\%$, $\eta_m = 95\%$, $\eta_{RWE} = 95\%$, rotor speed 1000 rpm, with design parameters ($\delta = 2$, $\beta = 5$, $\gamma = 0.75$). It is of interest to estimate the specific energy consumption and the size of the RWE device. The specific energy consumption can be directly obtained from Eq. (18), such that

$$\mathrm{SEC} = 1.45\left[\frac{1}{0.8} + \frac{1}{0.85}\left(\frac{1}{0.4} - 1\right)\right.$$
$$\left.(1 - 0.95 \times 0.95) + 1.5(0.4)^2\right] = 2.41\mathrm{KWh/m^3}$$

The reject brine is equal the brine inflow to the RWE become

$$Q_{\mathrm{Bi}} = \left(\frac{1}{R} - 1\right)Q_p = \left(\frac{1}{0.4} - 1\right) \times 50$$
$$= 75 \text{ m}^3/\text{h} \text{ or } 1.25 \text{ m}^3/\text{min}$$

The inner rotor radius can be obtained from Eq. (26)

$$R_I = \left[\frac{(Q_{\mathrm{Bi}}/N)}{0.5\pi\left(\frac{\beta}{\gamma}\right)}\right]^{1/3} = \left[\frac{(1.25/1000)}{0.5 \times 3.14 \times \left(\frac{5}{0.75}\right)}\right]^{1/3} = 0.05 \text{ m}$$

Hence,

$$R_o = 0.08 \text{ m}, \; S_H = 0.25 \text{ m}, \text{ and } L = 0.5 \text{ m}$$

According to the above results, it is possible to double both capacity to 100 m^3/h and rotor speed to 2000 rpm while maintaining the same overall dimensions.

7. Conclusions

The rotary work exchanger, RWE, concept is an improved PX device that can further reduce the cost of water by providing a less costly yet efficient means of energy recovery. One important feature of the RWE includes the flexibility of capacity variation through hydraulic external control means of rotor speed. Also, the study outlined the possible modes of operation of the RWE device, the various efficiency measures used in RWE, and the design consideration in sizing the RWE device. Moreover, a parametric study is performed illustrating the effect of efficiency of the high-pressure pump, booster pump, membrane and RWE on the specific energy consumption of a SWRO plant. The next step will be to build and test a prototype design.

Symbols

A_e	Effective channel cross-sectional area through the RWE
g	Gravity constant, m/s^2
H_{Bi}	Total inlet head of high-pressure brine to RWE, m.
H_{Bo}	Total outlet head of brine from RWE, m.
H_{Fi}	Total head of inlet seawater feed to RWE, m.
H_{Fo}	Total head of pressurized seawater feed to RWE, m.
h_{mi}	Head loss in inlet manifold, m
h_{mo}	Head loss in outlet manifold, m
h_r	Head loss in rotor channels, m
k_1, k_2	Empirical constants
L	Rotor length, m
N	Rotor speed, rpm
P_{Bi}	Pressure of brine inflow to RWE, kpa
P_{Bo}	Pressure of brine outflow from RWE, kpa
P_{Fi}	Pressure of seawater inflow to RWE, kpa
P_{Fo}	Pressure of seawater outflow from RWE, kpa
P_{MF}	Pressure of seawater feed to membrane, kpa
Q_{Bi}	Brine inflow to RWE, m^3/h
Q_{Bo}	Brine outflow from RWE, m^3/h
Q_{Fi}	Seawater inflow to RWE, m^3/h
Q_{Fo}	Seawater outflow from RWE, m^3/h
Q_{MF}	Feed flow rate to membrane, m^3/h
Q_p	Permeate flow rate, m^3/h
q_L	Internal leakage flow rate in RWE, m^3/h
R	Recovery ratio
S_H	Stroke length of high-pressure brine in RWE, m
X_{Bi}	Concentration of brine inflow to RWE, ppm
X_{Bo}	Concentration of brine outflow from RWE, ppm
X_{Fi}	Concentration of Seawater inflow to RWE, ppm
X_{Fo}	Concentration of Seawater outflow from RWE, ppm

Greek

β	Ratio of high-pressure stroke length to rotor inner radius
δ	Ratio of rotor length to high-pressure stroke length
γ	Cross-sectional area ratio of inlet header to rotor channel
η_{BP}	Efficiency of booster pump
η_{HP}	Efficiency of high-pressure pump
η_m	Efficiency of membrane
η_{pe}	Efficiency of pumping for SWRO
η_{RWE}	Efficiency of RWE
η_v	Volumetric efficiency of RWE
ρ	Seawater density, kg/m^3
ψ_d	Dilution factor, flow rate ratio of outlet brine to inlet brine

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