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Study of Efficiency in a Sliding Vane Pressure Exchanger

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Sliding vane pressure exchanger (SVPE) is expected to be an efficient device which recovers pressure energy from liquid streams in seawater reverse osmosis (SWRO) system. In this work, a matching design of vane number and port location was proposed to eliminate the short circuit flow, reversed flow, liquid decompression and compression. The contact performance between the cylinder and vane was studied by the proposed vane dynamics model. Then, the vane dynamics model in conjunction with the energy loss model were used to evaluate the work transfer efficiency of SVPE. The critical condition of contact performance and the work transfer efficiency of SVPE were simulated. Finally, the influences of device parameters on the work transfer efficiency were discussed. The results suggest a careful selection of device parameters, such as rotational speed, vane thickness and vane length, is critical in order to improve the work transfer efficiency of SVPE.

1. Introduction

The most common energy forms that can be recovered are heat and pressure. According to the developed heat exchange technology, the heat energy recovery potential is distributed in processes with temperature difference (Kilkovsky et al., 2015). Analogously, the pressure energy recovery can be achieved by pressure energy exchange between process streams with pressure difference. These streams exit in most pressure-driven process industries, such as seawater desalination industry.

Water scarcity is an increasing severe problem that hinders social development (Jiang, 2015). Seawater reverse osmosis (SWRO) technology is a signally efficient way in seawater desalination to solve this problem, but the high energy consumption blocks its development. Numerous efforts have been done to decrease the energy consumption, such as the configuration optimization of RO arrays, the improvement of the hydraulic permeability of RO membranes and the application of energy recovery devices (ERDs) (Altaee et al., 2016).

Presently, ERDs widely used in the SWRO system are based on the positive displacement principle including the piston-type work exchanger and rotary pressure exchanger. Due to the high work transfer efficiency of about 96.6 % (Stover et al., 2012), the rotary pressure exchanger has been a critical component to decrease the energy consumption (Kim et al., 2013). But meanwhile, the uncontrollable fluid mixing occurs within the device due to the direct contact between brine and seawater streams (Cao et al., 2015). The piston-type work exchanger is of low fluid mixing rate and high work transfer efficiency above 90 %. But its initial investment and maintenance cost are relatively high. Meanwhile, globe valves and servo valves are needed to precisely control the flow directions of fluid streams (Wang et al., 2016). In contrast to the positive displacement type ERDs, the centrifugal type ERDs, including Francis turbine, Pelton turbine, and turbochargers, have a maximum efficiency of about 82 % (Cameron et al., 2008). In recent years, a novel energy recovery device, termed 'sliding vane pressure exchanger (SVPE)', has been proposed (Al-Hawaj, 2011). The schematic of the SVPE is shown in Figure 1(a). The high-pressure brine stream pushes vane to drive rotor to rotate at brine side. The low-pressure seawater is pressurized by vane and flows into high pressure pipe network at seawater side. The energy is transferred from brine stream to seawater stream.

It is a pity that no more literatures about SVPE has been reported so far. In the reference (Al-Hawaj, 2011), models for flow rate, friction loss, and efficiency were presented under the assumption that each vane always contacts with cylinder. The reliable contact between the cylinder and vane is a requirement for the normal operation of SVPE, but the assumption of well contact performance has not been validated. Furthermore, an accurate method to precisely predict the work transfer efficiency is needed to provide detailed characteristics of SVPE. Hence, in this paper, the matching design of vane number and port location was proposed. The vane dynamic model was newly established based on the force analysis of the vane. Then, a method was introduced to determine the work transfer efficiency of SVPE based on the vane dynamics model and energy loss model. Finally, the influence factors of the work transfer efficiency and its effects were discussed.

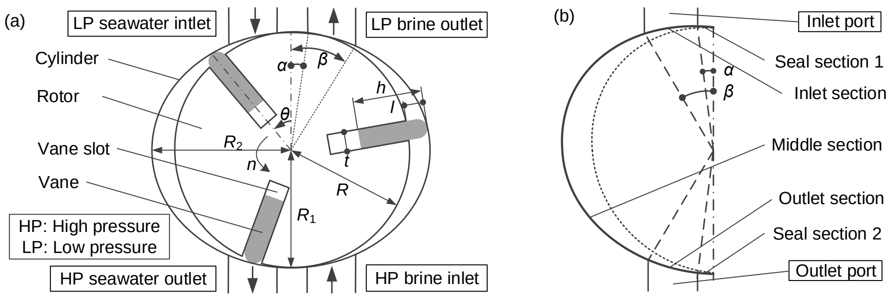

Figure 1: Schematic of (a) SVPE and (b) cylinder profile

2. Structural model

As shown in Figure 1(a), the rotor is concentrically disposed within the elliptical cylinder wall with a minor radius of R_1 and a major radius of R_2. The rotor radius is R which equals R_1. The vane is of t in thickness and h in length. The n is rotational speed, and θ is rotational angle. The N is vane number is, and l is the vane radius length out of the vane slot. The four ports are symmetrically arranged in cylinder with the port lower edge angular limit of α and port upper edge angular limit of β. As shown in Figure 1(b), the cylinder profile at brine or seawater side is divided into seal section 1, inlet section, middle section, outlet section and seal section 2 by α and β. And inlet section and outlet section connect to inlet port and outlet port.
The short circuit flow occurs with no vanes located in the middle section at brine side, as shown in Figure 2(a). With no vanes located in the middle section at seawater side, the reversed flow occurs due to the high pressure differential, as shown in Figure 2(b). If more than one vane is located in the middle section, the volume between the two adjacent vanes V_b increases and then decreases when the vane sweeps through the middle section, which results in the energy loss in the decompression and compression process of liquid, as shown in Figure 2(c). So, only one vane should be always located in the middle section. Based on the analysis above, the vane number matched with the port location is proposed to eliminate the short circuit flow, reversed flow, liquid decompression and compression, and a case of the matching design is presented in Table 1.

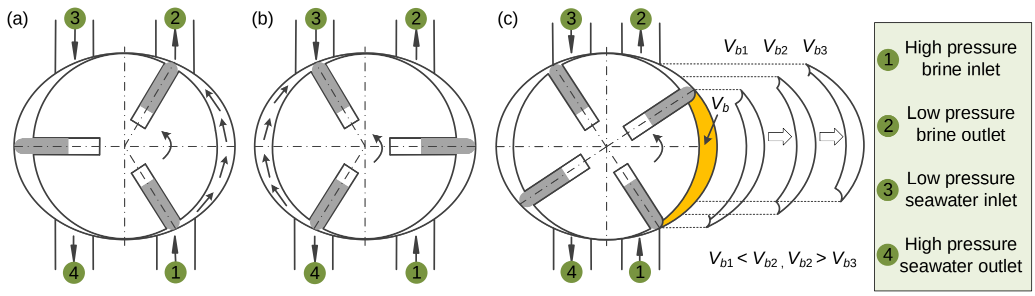

Figure 2: Schematic of (a) short circuit flow, (b) reversed flow, and (c) liquid decompression and compression

Table 1: A case of the matching design of vane number and port location

Vane number	Port lower edge angular limit / °	Port upper edge angular limit / °
3	5	30

3. Method for Determining Work transfer efficiency

3.1 Vane Dynamics Model

Forces acting on the vane whether the vane contact with cylinder or not should be considered separately, as shown in Figure 3. In two cases, the vane bears the gravitational force F_g, the inertial centrifugal force F_r, the inertial force of convected motion F_e, the coriolis inertial force F_k, the liquid force F_b acting on the vane bottom due to the liquid pressure in vane slot, the liquid force F_p acting on the vane side due to the liquid pressure difference between two sides of vane, the contact forces F_{n1} and F_{n2} at the two sides of the vane, the friction forces F_{f1} and F_{f2} at the two sides of the vane. In contact case, the vane tip divides liquid force into F_{pt1} and F_{pt2}. The vane is exerted the contact forces F_{nt} and friction force F_{ft} by cylinder. In non-contact case, the pressure within the neighbouring two chambers are the same, and the vane is exerted by liquid force F_{pt}. The inertia forces, F_e, F_r, and F_k, are obtained according to vane kinematics. The forces, F_p, F_b, F_{pt}, F_{pt1}, and F_{pt2}, are related to the liquid pressure at two sides of the vane and in the vane slot at given angle. The forces, F_{n1}, F_{n2}, F_{nt}, F_{f1}, F_{f2}, and F_{ft}, can be obtained by solving equilibrium equations.

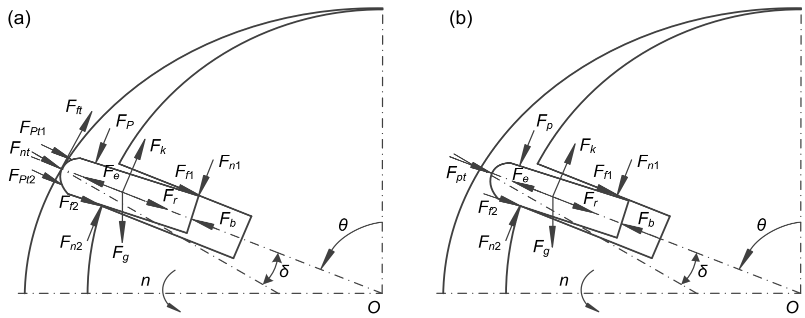

Figure 3: Forces acting on vane in (a) contact case and (b) non-contact case

In contact case shown in Figure 3(a), the force equilibrium equation can be written as:

$$\begin{pmatrix} -\cos\delta - \mu_1\sin\delta & -\mu_2 & -\mu_2 \\ \sin\delta - \mu_1\cos\delta & 1 & -1 \\ (\sin\delta - \mu_1\cos\delta)h/2 & -h/2 - \mu_2 t/2 & (l - h/2) + \mu_2 t/2 \end{pmatrix} \begin{pmatrix} F_{nt} \\ F_{n1} \\ F_{n2} \end{pmatrix} = \begin{pmatrix} F_r - F_e + F_g\sin\theta - F_b + F_{pt1} + F_{pt2} \\ F_k + F_g\cos\theta - F_p \\ F_p(l - h)/2 \end{pmatrix} \tag{1}$$

where δ is the inclined angle between vane axis and the normal line at the contact point in cylinder wall, μ_1 is vane tip friction coefficient, μ_2 is vane side friction coefficient.
In non-contact case shown in Figure 3(b), the contact forces at the vane sides are obtained respectively by:

$$F_{n1} = \frac{\left(F_r - F_e + F_g\sin\theta - F_b + F_{pt}\right)t/2 + \left(F_k + F_g\cos\theta\right)\left(-h/2 + l\right) - F_p l/2}{l - h - \mu_2 t} \tag{2}$$

$$F_{n2} = F_{n1} - F_k - F_g\cos\theta + F_p \tag{3}$$

The contact status between the cylinder and vane has a vital impact on the energy recovery process. And it is characterized by the value of the contact force F_{nt}. The vane should always contact with cylinder in the middle section, which means that the positive value of contact force F_{nt} in the middle section should be ensured.

3.2 Energy loss model

Relying on the smooth transition design of the cylinder inner wall and the precision machining of the device, the research in this paper is based on neglecting internal leakage. The brine and seawater streams have equal flow rates, and the work transfer efficiency η of SVPE is define as:

$$\eta = \frac{(p_{so} - p_{si})q_s}{(p_{bi} - p_{bo})q_b} = \frac{p_{so} - p_{si}}{p_{bi} - p_{bo}} \tag{4}$$

where p_{bi} is brine inlet pressure, p_{bo} is brine outlet pressure, p_{si} is seawater inlet pressure, p_{so} is seawater outlet pressure, q_b is the flow rate of brine stream, q_s is the flow rate of seawater stream.
The friction torque at vane tip and vane sides are obtained by:

$$M_{vt}(\theta) = \mu_1 F_{nt}\cos\delta\rho(\theta) \tag{5}$$

$$M_{vs}(\theta) = \frac{\mu_2}{\omega}\left|F_{n1}\frac{\mathrm{d}\rho(\theta)}{\mathrm{d}\theta}\right| + \frac{\mu_2}{\omega}\left|F_{n2}\frac{\mathrm{d}\rho(\theta)}{\mathrm{d}\theta}\right| \tag{6}$$

where $M_{vt}(\theta)$ is vane tip friction torque, $M_{vs}(\theta)$ is vane side friction torque, $\rho(\theta)$ is radial coordinate of cylinder profile, ω is angular velocity and equals $n\pi/30$. The friction torque at rotor end face and rotor side face are caused by the viscous drag of fluid which can be assumed as Couette flow, and can be expressed as:

$$M_{re}=\pi\mu\omega(R^4 - R_{sh}^4)/c_{re} \tag{7}$$

$$M_{rs}=4\pi\mu\omega R^3 b/c_{rs} \tag{8}$$

where M_{re} is rotor end face friction torque, M_{rs} is rotor side face friction torque, μ is dynamic viscosity of fluid, R_{sh} is shaft radius, c_{re} is end face clearance, c_{rs} is rotor side clearance. The rotor side clearance is considered to be the average value of the maximum clearance and minimum clearance, and equals $(R_2-R)/2$.
Then, the total friction power loss W_f can be written by:

$$W_f = \frac{N\omega}{2\pi}\int_0^{2\pi} M_{vt}(\theta) + \frac{N\omega}{2\pi}\int_0^{2\pi} M_{vs}(\theta) + \omega M_{re} + \omega M_{rs} \tag{9}$$

The friction power loss is considered to be the energy loss in the energy recovery process, and the calculated value of the work transfer efficiency based on the energy loss can be expressed by:

$$\eta_{cal} = 1 - \frac{W_f}{(p_{bi} - p_{bo})q_b} \tag{10}$$

3.3 Flowchart to determine the efficiency

Figure 4 shows the flowchart to determine the work transfer efficiency of SVPE. With an assumed value of the work transfer efficiency η, the seawater outlet pressure p_{so} is determined according to Eq(4). Then, the vane dynamics model is used to evaluate the contact performance between the cylinder and vane. The device parameters are adjusted if the vane cannot always contact with the cylinder in the middle section. Then, the energy loss model is utilized to calculate the energy loss in the energy recovery process. The calculated value of work transfer efficiency η_{cal} is obtained according to Eq(10), and then is used to update the assumed value of work transfer efficiency until the convergence of the computational process is achieved.

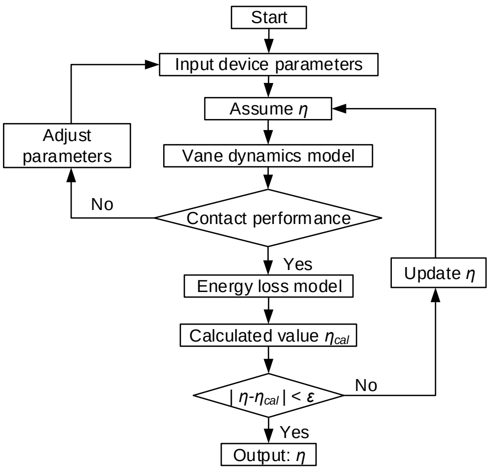

Figure 4: Flowchart for determining work transfer efficiency

4. Results and discussion

4.1 Contact performance

The critical parameters are used to define the critical contact performance that the vane can nicely always contact with the cylinder in the middle section. Figure 5 displays the critical rotational speed n_c varying with vane thickness and vane length. It can be observed from the figure that the critical rotational speed decreases with vane thickness and vane length respectively. The calculation results also show that the critical thickness decreases with rotational speed and vane length, and the critical length decreases with rotational speed and vane thickness. In other words, increasing rotational speed, vane thickness and vane length are all beneficial to the well contact performance.

4.2 Work transfer efficiency

Under the premise of well contact performance, the effect of rotational speed, vane thickness and vane length on the work transfer efficiency are simulated by the developed method. Figures 6 – 8 show the efficiency varying with rotational speed, vane thickness and vane length respectively. It can be seen from Figure 6 that the efficiency decreases with rotational speed since higher speeds means higher friction loss. According to Figure 7, the efficiency decreases with vane thickness. Larger vane thickness is associated with larger centrifugal forces due to the increase of vane mass, and also leads to larger vane base force due to the increase in vane cross-sectional area. Both effects result in increasing contact force and friction loss in vane tip. As shown in Figure 8, the efficiency slightly decreases with vane length. Increasing vane length results in larger centrifugal forces, leading to larger vane tip friction forces. But meanwhile, the vane centre is closer to the rotor centre, which improves the stress conditions of the vane. In general, increasing rotational speed, vane thickness and vane length have negative effects on the efficiency. According to Figures 6 – 8, the efficiency ranges 88.0 % – 96.4 %, and the maximum efficiency can achieve 96.4 % by calculation when the rotational speed, vane thickness and vane length are 2,000 r/min, 8.42 mm and 50 mm.

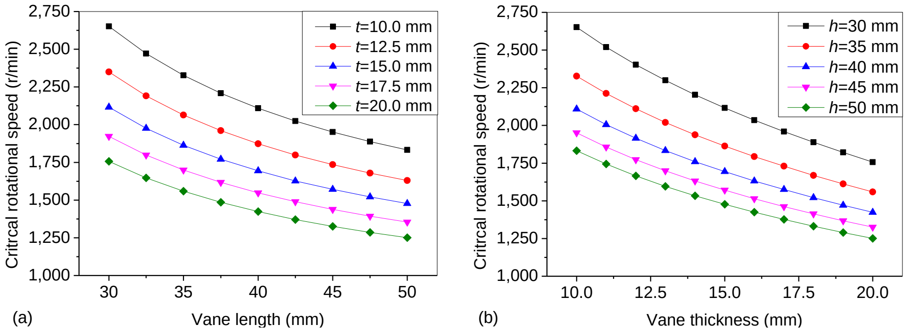

Figure 5: Critical rotational speed varying with (a) vane thickness, (b) vane length

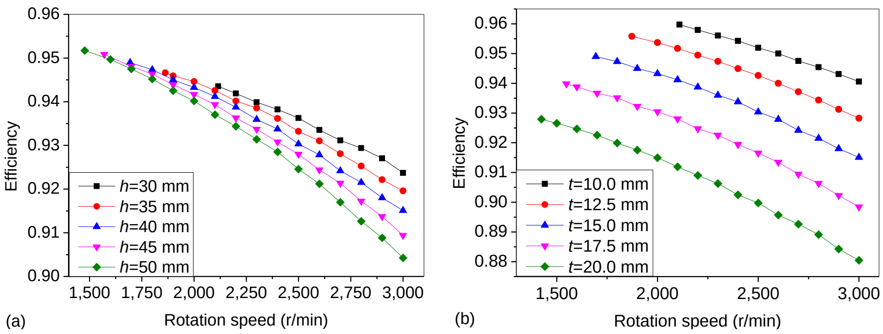

Figure 6: Effect of rotational speed on efficiency with (a) vane thickness of 15 mm, (b) vane length of 40 mm

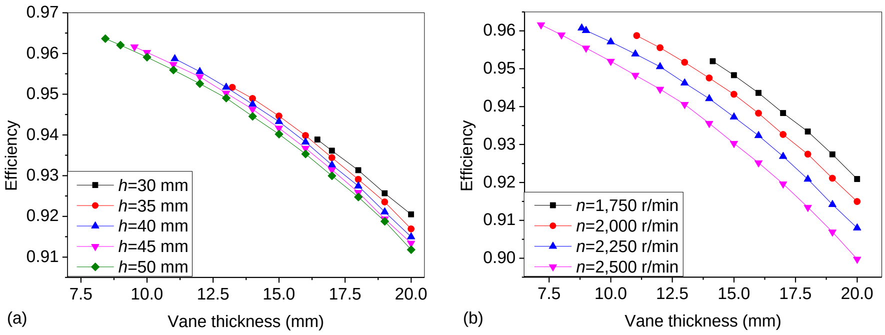

Figure 7: Effect of vane thickness on efficiency (a) rotational speed of 2,000 r/min,(b) vane length of 40 mm

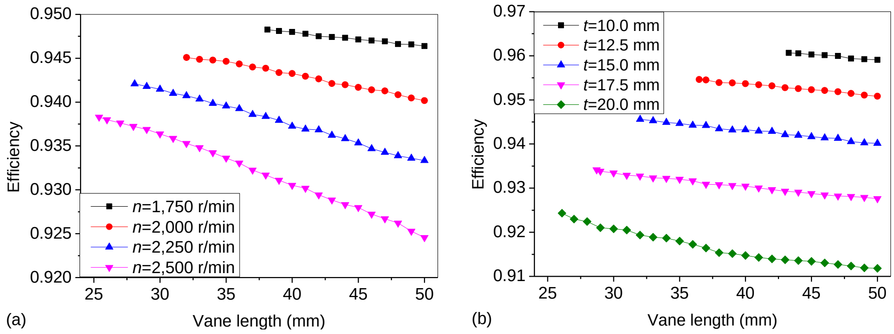

Figure 8: Effect of vane length on efficiency with (a) vane thickness of 15 mm and (b) rotational speed of 2,000 r/min

5. Conclusions

A method is developed to accurately evaluate the work transfer efficiency of a sliding vane pressure exchanger based on the mass and energy conservation. The three-vane SVPE matched with the port edge angles are designed and used to carry out following analysis. Decreasing rotational speed, vane thickness and vane length are all beneficial to achieve higher work transfer efficiency, but not conducive to the well contact performance between the cylinder and vane. The work transfer efficiency of SVPE ranges from 88.0 % to 96.4 % under different device parameters, and the maximum efficiency can achieve 96.4 % by calculation which approximates the efficiency of the positive displacement type ERDs. The results suggests that the SVPE is of reasonable structure, convenient manufacture and high efficiency, and expected to be a new type of efficient pressure energy recovery device. This work may provide a vital method to accurately predict the work transfer efficiency of SVPE, and help to guide performance optimization.

Acknowledgments

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