# Energy Efficiency Considerations for RO Plants: A Comparative Study

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## Abstract

One of the most important design features of an RO seawater plant is the specific energy consumption of produced drinking water. Specific energy consumption is largely dominated by two factors: the required transmembrane pressure at various mass transfer conditions, and the design and efficiency of the feed water pump in combination with the respective energy recovery system installed.

In this paper, the impact of various energy recovery strategies for single-stage RO-systems is evaluated and compared in terms of energetic efficiency. Comparative calculations, based on a real-case application, show that there is no such thing as the optimum configuration in general. Particular attention is paid towards the fact that each component has an optimum point of operation (flow and pressure), but in reality is hardly ever operated there. Since non-ideal operating parameters cause a deviation from optimum component performance, the combined interactive effects of two energy conversion devices (pump and recovery unit) operating under non-ideal, yet realistic, conditions are investigated. The purpose of this paper is to provide an overview regarding possible energy recovery strategies, and a comprehensive guideline on what aspects should be taken into account in order to optimize energetic performance of RO plants under real-life operating conditions.

The paper concludes that by using a novel throttling control scheme and by using existing energy recovery turbines in novel ways, significant improvements in energetic efficiency are possible compared with the current practice.

## 1. INTRODUCTION

Seawater desalination plants using Reverse Osmosis nowadays are considered to be state of the art. Major applications are for drinking water and water for industrial and irrigation purposes. From an operational and economical point of view, one of the most important design features of an RO seawater plant is the net feed pump energy consumption per unit volume of permeate produced (i.e. specific energy consumption). Specific energy consumption 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 installed in order to recover the available hydraulic energy in the discharged brine.

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. Yet even for given feed water properties at a given time, the required feed pressure will still depend on the current mass transfer conditions in the membrane system itself. Compared to membranes that have been recently cleaned, membranes that have been in operation for a period of time may give a significantly higher axial, i.e. feed-to-brine, pressure loss over the array due to colloidal fouling, thus requiring a higher feed pressure. As another aspect, scaling on back-end membrane elements will decrease specific water permeation, which again has to be compensated by higher feed pressures. Finally, 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.

Thus real-life operation of an RO plant will always be determined by a combination of the above mentioned factors. Even if flow rates are kept constant by means of operating the plant at a constant-recovery mode, there is no fixed duty point. Instead, there will be a duty range, with the width of this range being governed by seasonal changes in feed water properties as well as the cleaning strategies applied by the operator. In contrast to this, the hydraulic devices, i.e. feed pump (typically centrifugal) and energy recovery device (typically turbine), 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. As a consequence of this contradiction between duty range determined by the membrane/water system and the given duty point of the hardware, energy will be wasted due to non-optimum pump and recovery turbine operation.

Therefore, the key challenge in plant design is to find a hydraulic configuration that provides the minimum integral (i.e. year-round average) specific energy consumption. Basically, this means finding a compromise between the required operational range and the hydraulic characteristics of pump and recovery turbine. In this paper, several configurations are investigated to determine their respective impact on energy consumption for a design of a representative RO plant. The purpose of this analysis is to provide a comprehensive guideline on what aspects should be considered to optimize energetic performance of RO plants under real-life operating conditions.

## 2. BASIC ASSUMPTIONS

The calculations in this paper are based on a case study for an RO desalination project in the Arabian Gulf producing water for irrigation purposes. Table 1 shows the relevant data for year-round feed water conditions as determined from on-site samples. These conditions have been used for all the model calculations. In terms of salinity and temperature, this analysis can be considered representative for seawater conditions in the Arabian Gulf. Feed pressure values given in the table are taken from RO design software under the assumption that the plant is designed to operate at a constant recovery rate of 40 %.

| Average month | Salinity [mg/l] | Spec. Gravity [kg/m3] | Minimum Temperature [°C] | Maximum Temperature [°C] | Feed Pressure @ min. Temp. [bar] | Feed Pressure @ max. Temp. [bar] |
|---|---|---|---|---|---|---|
| January | 45,10 | 1045,1 | 18 | 22 | 81,5 | 77 |
| February | 45,8 | 1045,8 | 18 | 22 | 81,2 | 78,3 |
| March | 45,7 | 1045,7 | 20 | 24 | 80,5 | 76,0 |
| April | 45,94 | 1045,9 | 24 | 30 | 76,6 | 71,9 |
| May | 45,84 | 1045,8 | 28 | 33 | 73,1 | 69,9 |
| June | 46,47 | 1046,5 | 32 | 34 | 71,6 | 70,4 |
| July | 46,37 | 1046,4 | 34 | 36 | 70,2 | 69,3 |
| August | 44,86 | 1044,9 | 34 | 36 | 67,5 | 66,5 |
| September | 44,78 | 1044,8 | 32 | 35 | 68,3 | 66,8 |
| October | 45,18 | 1045,2 | 26 | 30 | 73,2 | 70,4 |
| November | 45,24 | 1045,2 | 20 | 26 | 79,5 | 73,4 |
| December | 45,2 | 1045,2 | 18 | 22 | 82,1 | 77,1 |

Table 1: Arabian Gulf feed water properties and corresponding feed pressure requirements

In order to account for the varying membrane conditions that will occur during operation, a total pressure loss over membrane array and manifold system of 2 bar (clean membranes) and 6 bar (fouled membranes) have been assumed. For the feed pump, it has been assumed that the respective duty point coincides with the given feed flow rate at the maximum differential pressure of 80.1 bar (membrane pressure for December at minimum feed temperature minus 2.0 bar pump inlet pressure). Thus the feed pump duty point determines the gross amount of energy introduced into the system, whereas the amount of recovered energy will be determined by the various recovery conditions. The hydraulic assumptions are summarized in Table 2.

| | |
|---|---|
| Feed water properties | acc. to Table 1 |
| Feed flow rate | 950 m3/h |
| Permeate recovery | 40 % |
| Motor efficiency | 98 % |
| Feed pump inlet pressure | 2 bar |
| Feed pump efficiency | 85 % |
| Feed pump type | centrifugal |
| Minimum array/manifold loss | 2 bar |
| Maximum array/manifold loss | 6 bar |

Table 2: Hydraulic assumptions for model calculations

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

## 3. CONTROL BY FEED THROTTLING, RECOVERY BY PELTON TURBINE

One of the simplest methods to operate the 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 turbine as depicted in Figure 2. Permeate backpressure is 0 bar. In this case, the Pelton turbine will be operated at a constant flow rate due to the constant recovery, but inlet pressures will differ in accordance with pressure losses on the feed side. The efficiency curves of the turbines considered are given in Figure 1; turbine discharge pressure is of course 0 bar in all cases.

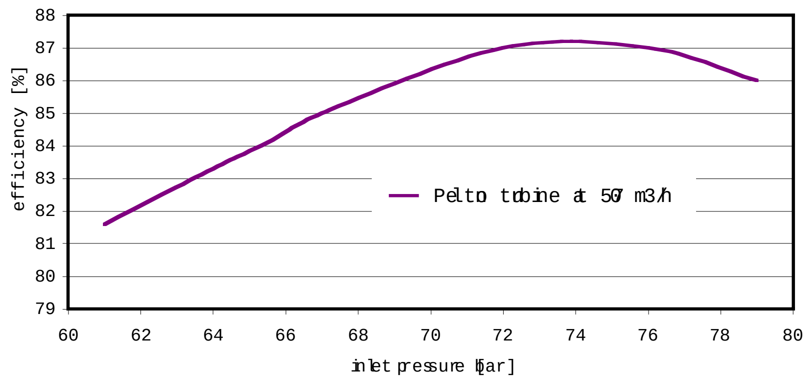



Figure 1: Pelton turbine characteristics

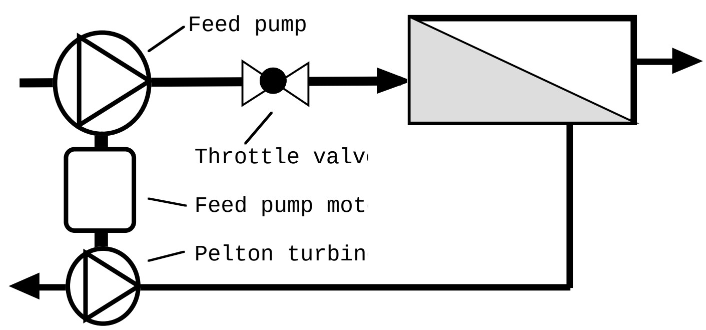



Figure 2: Feed throttling control, Pelton turbine energy recovery

The following four cases define the operating extremes:

- case 1: maximum array loss at minimum temperature
- case 2: minimum array loss at minimum temperature
- case 3: maximum array loss at maximum temperature
- case 4: minimum array loss at maximum temperature.

Figure 3 shows the energy consumption per cubic meter of permeate produced in the plant under the assumptions listed in Tables 1 and 2. It contains several interesting conclusions. Specific energy consumption reaches its maximum value during the months where the required feed pressure is at a minimum due to the high temperature. This seems contradictory at first glance, but of course

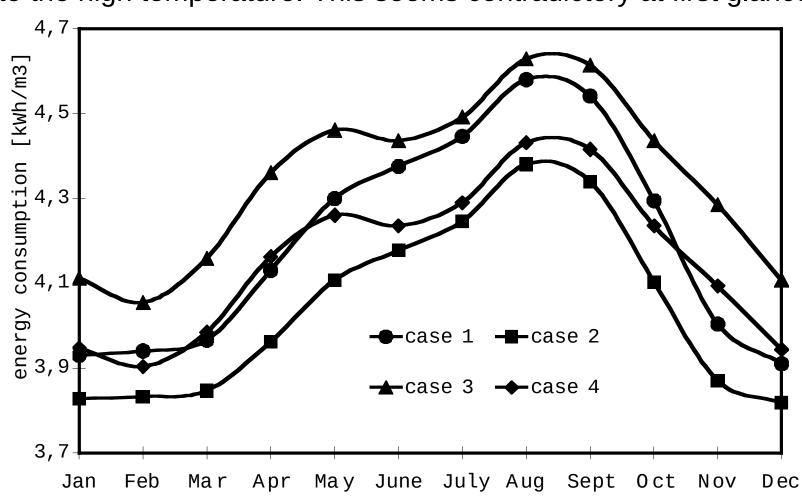



Figure 3: Specific energy rate for feed throttling control, Pelton turbine recovery

the reason is the feed control valve. Since the pump is always operated at maximum pressure, the feed control valve has to dissipate a high amount of energy in order to obtain the required feed pressure resulting in a reduced pressure at the turbine inlet. This effect is amplified by the turbine efficiency curve, because throttle losses cause a deviation to the left, i.e. towards lower inlet pressures and thus a decreased efficiency in the off-duty range. Fouled membranes account for a 0.2 kWh/m3 increase in specific energy consumption, which represents an approximate increase of 5%. This figure should be taken into account when membrane cleaning strategies are being discussed.

## 4. 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 Pelton turbine. The process diagram is shown in Figure 4. In this case, no throttling occurs on the feed side because

permeate backpressure is increased instead in order to decrease transmembrane pressure difference. Again, the Pelton turbine will be operated at a constant flow rate due to the constant recovery, but inlet pressure variations will be less pronounced and only be determined by the array losses.

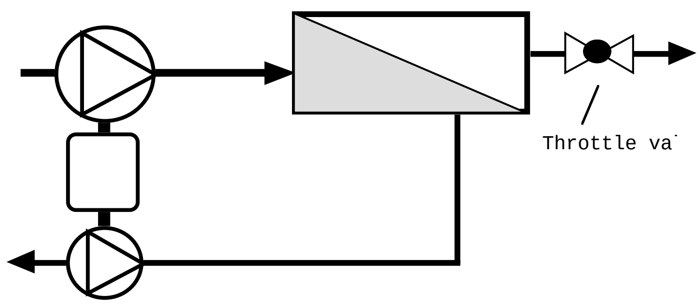



Figure 4: Permeate throttling control, Pelton turbine energy recovery

Interesting results are shown in Figure 5. In this case, the temperature has no effect on the energy consumption. Instead, it is constant throughout the year. The reason is that the turbine inlet pressure is constant except for the decrease caused by membrane fouling. Since the throttling occurs on the permeate side, only 40% of the energy is dissipated when compared to feed throttling. Consequently, the turbine can recover more of the energy introduced by the pump, because it operates on a higher average pressure level, and closer to the design duty point. The latter effect also accounts for a smaller difference between clean and fouled membranes. As a general conclusion, permeate throttling is significantly more energy efficient than feed throttling.

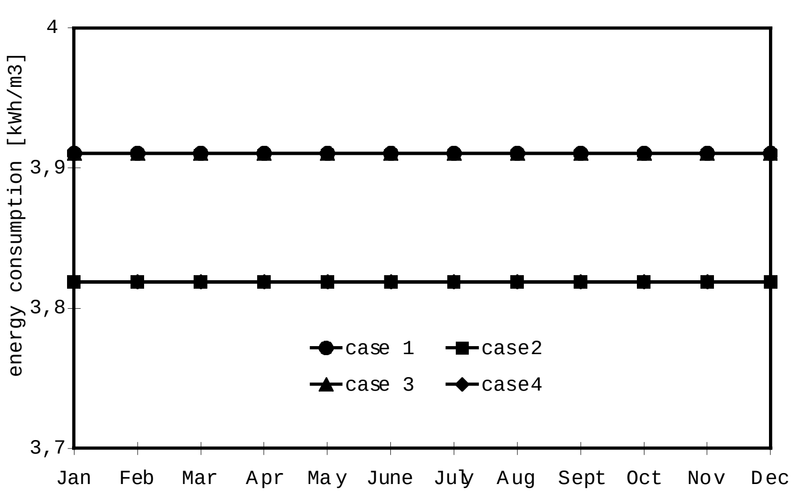


Figure 5: Specific energy consumption for permeate throttling control, Pelton turbine energy recovery

## 5. CONTROL BY VSD ON FEED PUMP, RECOVERY BY PELTON TURBINE

The configuration in this case is basically the same as shown in Figure 2, except that the throttle valve is deleted and the feed pump motor is driven by a frequency converter (Variable Speed Drive, VSD). In the calculations, VSD conversion losses have been accounted for by assuming a converter efficiency of 98%. Results for this process are given in Figure 6. The VSD provides energy savings by reducing pump speed (hence pressure and energy consumption) as required to meet the reduced transmembrane pressure difference at higher temperatures.

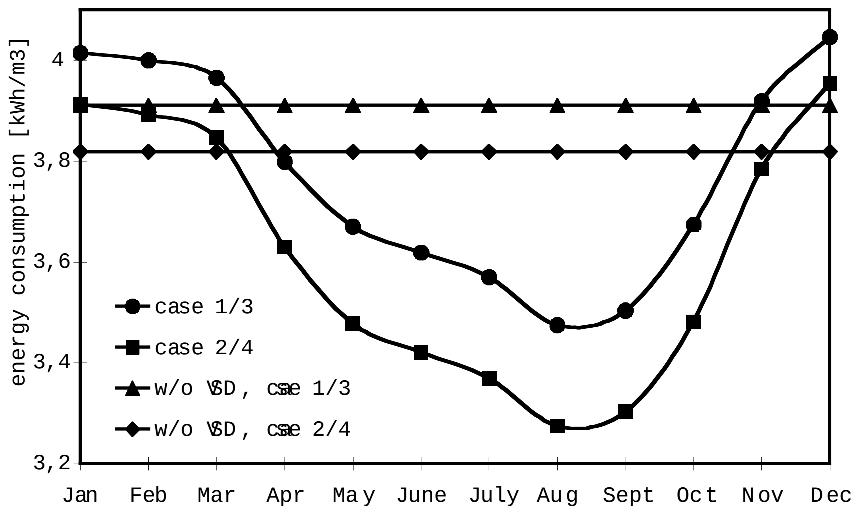



Figure 6: Comparison of specific energy consumption for permeate throttling and VSD controls, Pelton turbine energy recovery

Figure 6 also compares the process with a VSD to the same process without a VSD on the pump but with permeate throttling as outlined in Section 4. Note that in case of cold temperatures, specific energy consumption is higher when using the VSD. Two reasons cause this effect: VSD conversion losses, and also the larger deviation from the turbine duty point towards lower pressures.

The question whether the additional investment for the VSD pays back the savings in energy consumption very much depends on the energy costs, but also on depreciation details for the specific project. The general statement is: The higher the costs of energy, the more valuable a VSD becomes. Model calculations indicate that the break-even for the use of a VSD is approximately above 0.035 US-$ per kWh for the temperature characteristics used in this analysis. Of course an application with smaller temperature variations will increase this break-even, since the savings potential is less.

So far, the design of the plant included a feed flow rate and feed pressure that matched the optimum duty point of the pump and the turbine, which is one of the basic design rules. The effect of a poor hydraulic match between the pump and recovery turbine with respect to the membrane design pressure requirement can be substantial. Assume for the moment, that the feed pump efficiency were 81% instead of 85% and the turbine efficiency were reduced by 2% over its performance range due to off-design operation. Even with a VSD, overall energy consumption of the off-design system exceeds what would have been required in case of a proper design (and no VSD). Since there is more variety in terms of plant hydraulics than in the hydraulic characteristics of pumps and turbines available on the market, RO plant design should be guided by the available equipment, instead of forcing existing equipment into plant hydraulics that are often arbitrarily determined. In other words, do not force the pump and turbine to meet the requirement for a 10,000 m3/day target capacity if the specific energy rate could be substantially less by operating at, say, 10,800 m3/day.

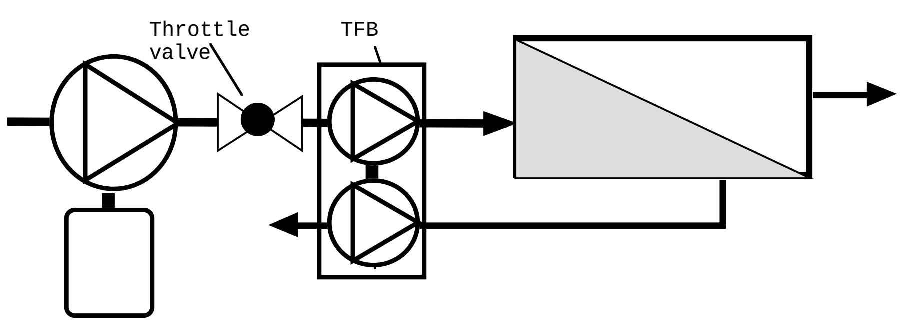
## 6. CONTROL BY THROTTLING, RECOVERY BY TURBOCHARGER

The turbocharger feed booster (TFB) is an integral feed pump (single stage centrifugal) and energy recovery turbine (single stage radial inflow). TFBs from several manufacturers have been in wide use since the early 1990s. The TFB has two interesting characteristics: 1) it returns recovered brine energy to the RO system in the form of a pressure boost in the feed stream and 2) it has a variable area nozzle that permits adjustment brine flow through the turbine resulting in an adjustable pressure boost in the feed stream. This combination of characteristics opens the possibility of using the TFB as a pressure-leveling device in an RO system. That is, the TFB brine flow and/or brine pressure 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).

A third unique characteristic of the TFB is that it reduces the size of the feed pump and feed pump motor by reducing the feed pump discharge pressure. This provides a potentially substantial reduction in capital costs. Also, at least for large systems, it opens the possibility of using single stage feed pumps, which may have a substantially lower cost than multistage pumps and may have a higher efficiency due to a simplified hydraulic flow path. However, for this analysis no efficiency advantage was granted to the feed pump used with the TFB.

TFB efficiency is defined as the ratio of hydraulic energy transferred to the feed stream to the hydraulic energy available in the brine stream (“hydraulic transfer efficiency”). A TFB manufacturer suggested a 73% transfer efficiency for the flow and pressure range in this analysis[a]. The manufacturer indicated that the TFB’s variable area nozzle could accommodate a 25% brine pressure range at constant brine flow, which would meet the brine pressure variation specified in this analysis. Note that the TFB efficiency includes the hydraulic losses in its integral feed pump section as well as its turbine. In comparison, Pelton turbine efficiency only includes turbine losses, which preclude direct comparison of the respective device efficiencies.

The TFB can control feed pressure two ways. Figure 7 illustrates using a feed throttle valve. Figure 8 illustrates bypassing brine through a small control valve. Bypassed brine reduces the energy available to the TFB while the TBF brine nozzle would be adjusted to maintain the desired brine



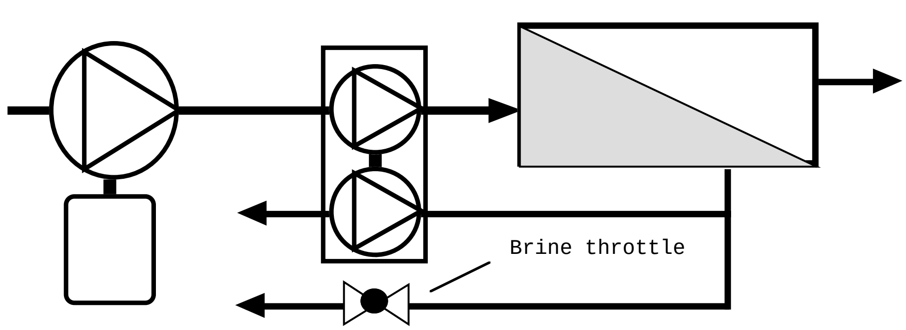



Figure 7: TFB with feed throttle valve (above). Figure 8: TFB with brine bypass (below)

pressure. There is no energetic advantage to brine bypassing relative to feed throttling but a brine bypass does replace the large feed throttling valve with a much smaller brine valve.

Figure 9 shows what may seem to be a surprising specific energy rate: The rate is unvarying regardless of feed pressure requirement or fouling-induced change in pressure loss along the membrane channel. The reason is that the feed pump sees only a constant flow, pressure differential and speed hence its energy consumption is constant. Energy recovery and feed pressure control occurs in equipment that is independent of the feed pump (i.e. the TFB and a control valve).

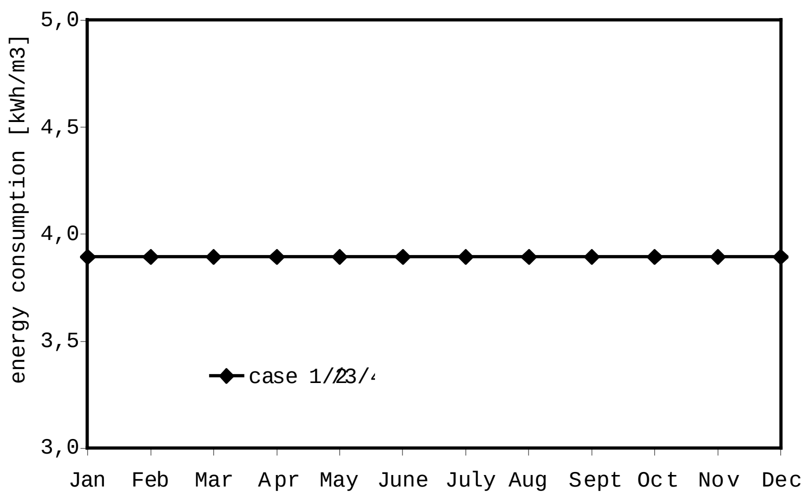



Figure 9: Specific energy consumption for TFB with feed throttling or brine bypass

## 7. CONTROL AND RECOVERY BY TFB AND HELPER TURBINE

The throttling process imposed on the bypass brine stream depicted in Figure 8 can be eliminated by directing the bypass brine though a “Helper turbine” which is connected to the high pressure feed pump shaft[b]. See Figure 10.

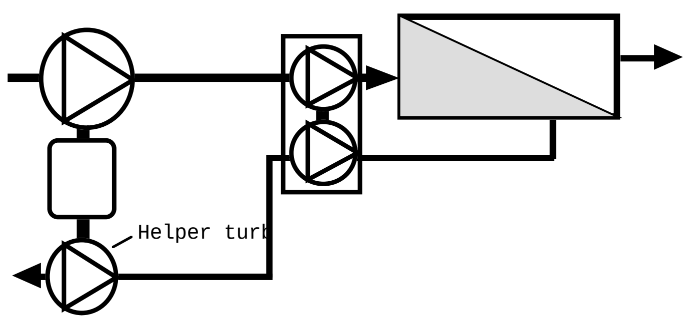



Figure 10: TFB with helper turbine in parallel with TFB turbine

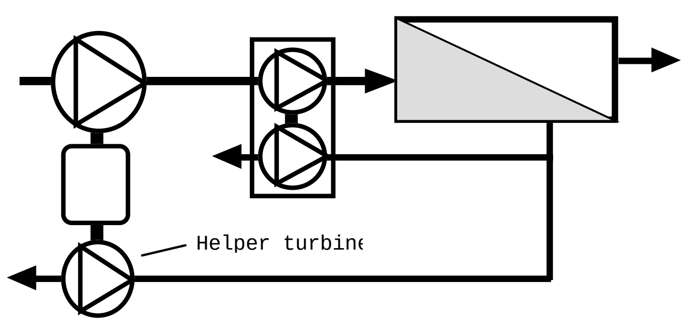


Figure 11: TFB with helper turbine in series with TFB turbine

The brine bypass volume will vary from zero (at maximum membrane pressure) to a substantial value (at minimum membrane pressure). Of course, the Helper turbine has a certain minimum flow rate for efficient operation and only when that minimum flow rate is available can the Helper turbine be engaged for energy recovery. Generally, a Pelton turbine would be the best choice for this application due to its flat efficiency over a broad flow range.

A variation of above mentioned design is to use an energy recovery turbine that is in series with the TFB as depicted in Figure 11. Here, the Helper turbine handles the entire brine flow but operates at a pressure differential that equals the total pressure differential of the brine stream minus the pressure differential across the TFB. With the TFB operating at a constant flow ratio, it does not suffer from off-design losses. Moreover, this arrangement reduces the potential for erosion/corrosion that may otherwise occur with high velocity brine streams; especially of concern with Pelton turbines.

The Helper turbine concept has the unique advantage of eliminating throttling losses for most of the operating cycle in the RO process without resorting to VSD equipment with the associated energy losses and high capital cost. Figure 12 shows the specific energy consumption of the TFB with Helper turbine along with the brine flow to the Helper turbine. The chief assumptions are that the minimum flow is 15% of the maximum flow and that the Helper turbine has an average efficiency of 85%. The index “Q” in the graph refers to the brine flow through the helper turbine.

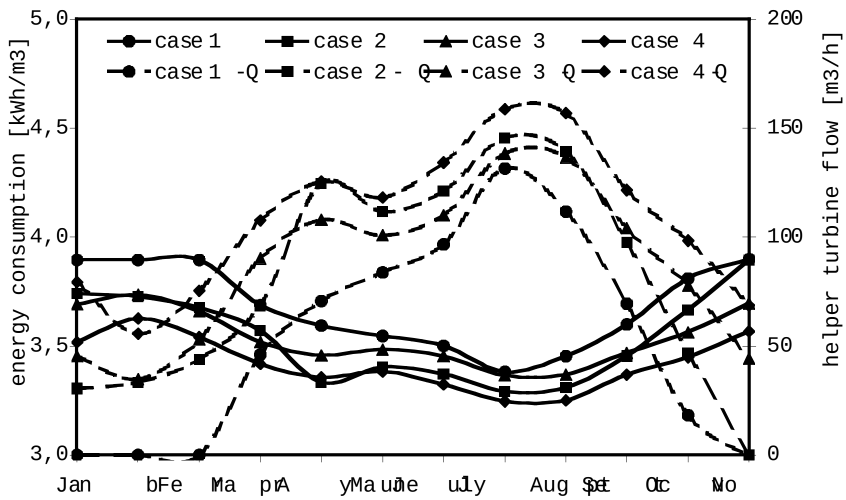



Figure 12: Specific energy consumption with TFB and Helper turbine and brine flow through the helper turbine

## 8. BRINE CONVERSION SYSTEM WITH TFB AND HELPER TURBINE

Perhaps the chief task in achieving high efficiency is to avoid throttling losses. Yet, the largest loss in RO systems is rarely addressed; the irreversible pressure loss that occurs across the membrane itself (i.e. transmembrane pressure). This loss can approach 50% of the feed pump energy input.

Ideally, the transmembrane pressure should remain constant along length of the membrane channel however the rapidly increasing osmotic pressure due to permeation and the reduction in pressure energy due to frictional losses conspire against this objective. Therefore, the designer is forced to accept undesirably high transmembrane pressure for the initial membrane elements simply to ensure that the transmembrane pressure is adequate for the last elements. Not only is this wasteful of feed pumping energy but also may accelerate fouling due to overfluxing of the initial elements.

The “Brine Conversion System”[c] (BCS) represents an attempt to minimize excessive transmembrane pressure by using a two-stage membrane configuration with the later stage operating at a higher pressure to compensate for the otherwise inevitable reduction in transmembrane pressure. The BCS is usually described as using a TFB to accomplish the interstage pressure boosting and possibly using a Helper turbine to utilize excess brine energy. See Figure 13.

If the BCS can be proven able to significantly reduce irreversible permeation throttle losses then the most energetically efficient system may be a BCS equipped with a TFB and Helper turbine. However, more research is needed in this area before conclusions can be drawn.

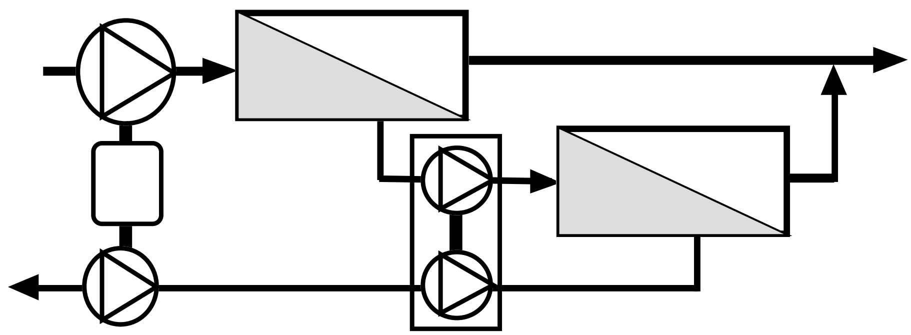


Figure 13: Interstage pressure boosting using TFB with Helper turbine

## 9. SUMMARY COMPARISONS

Figure 14 shows the specific energy consumption for perhaps the three “most conservative” approaches (i.e. lowest technical risk) for large system energy recovery considered in this report; Pelton turbine with feed throttling, TFB with throttling and TFB with Helper turbine. Each device’s energy rate has been averaged for each month over the four operating cases.

The TFB holds a slight average advantage over the Pelton turbine when both are used with throttling. This may seem surprising as the Pelton turbine seemingly has a higher efficiency. However, this fact provides an excellent example of the importance of evaluating the interaction of the feed pump, membrane and the turbine across the range of expected operating conditions.

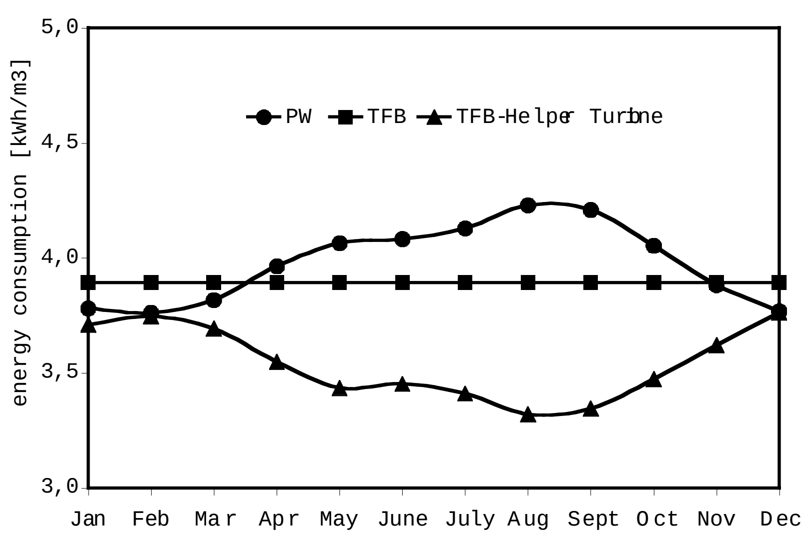



Figure 14: Comparison of specific energy consumption averaged over the four operating cases

The TFB with a Helper turbine is the most efficient of the approaches considered as it largely eliminates throttling losses (without resorting to an expensive VSD).

Table 3 summarizes the specific energy rate for all equipment configurations considered in this paper. The worst efficiency configuration is also perhaps the most widely used; the Pelton turbine with feed throttling. Table 3 also shows that the TFB with throttling reduces energy consumption by about 2.6%. Here equipment section may be decided by weighing the importance of capital cost (probably in favor of the TFB) and operating experience (probably in favor of the Pelton turbine in large systems).

Table 3 shows permeate throttling holds the potential for excellent efficiency. However, transient pressure reversals can damage the internal sealing of the membrane sheets. Permeate throttling should be considered with caution. Table 3 also shows that use of a VSD provides the expected efficiency advantage. However, a high capital cost and questions concerning reliability maintainability should be considered.

The combination of a TFB and Pelton turbine appears to be the most efficient of all. Basically, the TFB and Pelton turbine can synergistically interact via their unique hydraulic characteristics to provide independent control of feed pressure without throttling (via the TFB) and fully utilize brine hydraulic energy (Pelton turbine).

| **Device** | **Approx. Average Energy Rate (kWh/m3)** |
|---|---|
| **Pelton turbine with feed throttling** | **4.01** |
| **TFB with feed throttling or brine bypass** | **3.90** |
| **Pelton turbine with permeate throttling** | **3.85** |
| **Pelton turbine with VSD** | **3.70** |
| **TFB with helper turbine** | **3.54** |

Table 3: Overall average specific energy rates

## 10. CONCLUSIONS

Particular attention is paid towards the fact that each component has an optimum point of operation (flow and pressure), but in reality is hardly ever operated there. Since non-ideal operating parameters cause a deviation from optimum component performance, the combined interactive effects of two energy conversion devices (pump and recovery unit) operating under non-ideal, yet realistic, conditions are investigated.

One major reason for a deviation from the optimum operational parameters is the shift in mass transfer conditions over time due to fouling and/or scaling. Since worst case conditions have to be considered in order to properly size the pump as well as the energy recovery unit, energy dissipation in a throttle valve is often used to adapt to less-than-worse case conditions. In addition, from the viewpoint of the component designer and manufacturer, required RO feed flows and pressure often do not coincide with the optimum performance of the pump or turbine. It can be shown that a slight adjustment in the feed flow rate towards optimum pump and turbine performance can significantly increase the energetic efficiency. The benefits of a VSD driving the pump is investigated as an alternative solution and evaluated. The comparison of additional investment costs versus savings in operational costs shows that the installation of a VSD can often compensate for a non-ideal match between the given energy efficiency characteristics of the pump and the energy recovery device, and required operational parameters of the RO unit.

In view of the priority of eliminating throttling losses and the desirability of a constant speed pump drive, the best energy recovery devices appears to be the TFB with a helper turbine; preferably a Pelton type. The BCS with the TFB and Helper turbine is potentially the most efficient of all configurations considered in this paper if it can deliver on the promise of reducing the irreversible losses in the permeation process.

As a result of this study, recommendations for the optimum solution for maximum efficiency can be deduced based on the operational parameters required by the RO process.

## 11. REFERENCES

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