Hydraulic turbocharger for energy recycling in the reverse osmosis seawater desalination system

Yunguang Ji¹, Xiaoxia Li and Hongtao Li
School of Mechanical Engineering, Hebei University of Science and Technology, Shijiazhuang, Hebei 050018, China
¹Email: Jiyg@hebust.edu.cn

Abstract. In order to reduce the energy consumption of the reverse osmosis seawater desalination process, a hydraulic turbocharger is proposed to recycle the pressure energy of the brine. A high efficiency energy recovery hydraulic turbocharger is designed based on Computational Fluid Dynamic method and installed in the reverse osmosis seawater desalination system of Dagang Power Plant in Tianjin Development Zone, China. The actual operation results show that the hydraulic turbocharger has a pressure energy recovery efficiency of up to 72%, and with high working stability and low noise, which may provide a reliable reference for the design and application of pressure energy recovery devices.

1. Introduction
Desalination is considered to be the most practical and ideal method for continuously providing fresh water sources [1]. In order to reduce effectively the energy consumption of the desalination process, a variety of energy recovery devices are applied. The traditional commonly used energy recycling turbines have the problems of high failure rate and serious loss of efficiency [2].

As a new type of pressure energy recovery device, hydraulic turbocharger has obvious structural and efficiency advantages compared with existing hydraulic turbines [3]. As shown in Figure 1, the pump-side impeller and the turbine-side impeller adopt a coaxial design, and there is no over-extension shaft, no dynamic seal is required, and the rotating components are completely enclosed in the sealed case so as to achieve zero leakage. There are three bearings in the hydraulic turbocharger. The pump side bearing and the center bearing are radially supported, and the thrust bearing on the turbine side balances the axial force from the lateral side of the pump to the turbine side due to the liquid pressure difference [4, 5]. In addition, an auxiliary pipe is provided to regulate the flow rate to ensure that the hydraulic turbocharger works in the high efficiency range to better adapt to fluctuating operating conditions. Figure 2 shows the prototype of the hydraulic turbocharger.

The high-pressure brine of the reverse osmosis seawater desalination process drives the turbine-side impeller and then the shaft to rotate, hence the medium on the pump side can be pressurized by the pump impeller. During in this process, the pressure energy on the turbine side is recycled. Figure 3 illustrates the diagram of application of the hydraulic turbocharger in the reverse osmosis desalination system.

Taking the seawater desalination process system of Dagang Power Plant in Tianjin Development Zone as the example, the models on both sides of the hydraulic turbocharger are designed according to the process system parameters. The flow rate on the pump side is 150 m³/h, the pressure is required to
be 2 MPa, the flow rate on the turbine side is 90 m$^3$/h, and take 10% flow rate in the turbine side for regulating. The inlet pressure of the turbine is 4.6 MPa. The process parameters are listed in Table 1.

Table 1. Hydraulic turbocharger process parameters.

| Parameters                     | Value |
|-------------------------------|-------|
| Pump flow/Q$_P$, m$^3$/h      | 150   |
| Pump inlet pressure/P$_{P,in}$, MPa | 2.8  |
| Pump outlet pressure/P$_{P,out}$, MPa | 4.8  |
| Turbine flow/Q$_T$, m$^3$/h    | 90    |
| Turbine inlet pressure/P$_{T,in}$, MPa | 4.6  |
| Turbine outlet pressure/P$_{T,out}$, MPa | 0.1  |

2. Hydraulic model design and flow field simulation

2.1. Hydraulic model of turbine side

The turbine side of the turbocharger is modelled according to the conversion relationship between the pump and the hydraulic turbine [6].
In order to improve the efficiency of the hydraulic turbine and widen its high-efficiency flow range, the speed is selected to be 8000r/min. The main geometric parameters of the impeller after optimization of the turbine hydraulic model are shown in Table 2 according to the design parameters of the seawater desalination process system in Table 1.

| Parameters                       | Value |
|----------------------------------|-------|
| Impeller outlet diameter/D₁/mm  | 36    |
| Impeller inlet diameter/D₂/mm   | 180   |
| Impeller inlet angle /°          | 90    |
| Impeller exit angle /°           | 62.5  |
| Impeller wrap angle /°           | 40    |

2.1.1. Flow field simulation of turbine side. The fluid calculation domain consists of the volute and the impeller. The meshing results are shown in Figure 4.

2.1.2 Simulation results of turbine side. As shown in Figures 5 and 6, the pressure contour and velocity vector are simulated. The pressure gradient is uniform. The inlet and outlet pressures meet the design requirements of 4.6 MPa for inlet pressure and 0.1 MPa for outlet. There is no obvious eddy current phenomenon inside, which satisfies the design conditions.
2.2. Hydraulic model of pump side

The turbine side is connected to the pump side by the shaft and transmits energy. The torque output from the turbine side needs to be greater than the torque required by the pump side to achieve pressurization of the medium.

When the angular velocity is the same, the turbine side shaft power is greater than that of the pump side to achieve turbine boosting. According to the requirements of shaft power and seawater desalination process system, the pump side impeller geometry parameters are obtained, as shown in Table 3.

| Parameters                  | Value |
|-----------------------------|-------|
| Impeller outlet diameter/D₁/mm | 160   |
| Impeller inlet diameter/D₂/mm | 40    |
| Exit width/b₂/mm            | 14    |
| Impeller exit angle/β₂/°     | 22.5  |
| Number of blades/z           | 7     |

The pump side model still uses ANSYS-CFX16.0 to simulate the constant flow field of the impeller and volute of the pump side as done in turbine side. When the pump side model is completed, the outlet port on the turbine side is the inlet port on the pump side, and the inlet port on the turbine side is the outlet port on the pump side [7]. The pressure contour and velocity vector are shown in Figures 7 and 8. The maximum pressure on the pump side can reach up to 4.8 MPa, which meets the pressure increase requirement of 2 MPa.

3. Testing and application of hydraulic turbochargers

3.1. Efficiency and economic feasibility assessment of the hydraulic turbocharger

3.1.1. Energy recovery efficiency. It can be calculated by equation (3).

\[
\eta = \frac{Q_T (P_{p,out} - P_{p,in})}{Q_T (P_{T,in} - P_{T,out})}
\]  

(3)

The energy recovery efficiency of the hydraulic turbocharger of the intended design calculated by equation (3) is 74%.

3.1.2. Economic feasibility assessment. Take 3 years to recycle the equipment investment as a reference. The judging principle is:
\[ K = \left( \frac{M_1 \times B_{HP} \times 26280}{M_2} \right) \geq 1 \] 

In equation (4), \( K \) is the economic coefficient; \( B_{HP} \) is the output power, kW; \( M_1 \) is the electricity cost, yuan/kW; \( M_2 \) is the one-time investment of the hydraulic turbocharger for 3 years, and the estimated maintenance cost, yuan.

3.2. Testing system
The test system mainly consists of a multi-stage pump, an water tank, an air compressor, hydraulic turbocharger, pressure gauges and flow meters, control valves and pipes, etc.

The hydraulic turbocharger performance test is performed at the same speed but different flow rate. Figures 9 and 10 show the hydraulic turbocharger performance test system and its schematic diagram. The butterfly valve 3 is used to control the water switch of the entire system and needs to be opened during the test and closed after the test. 4 and 8 are electromagnetic flow meters. At the beginning of the test, the water in the water tank was pressurized by the multi-stage pump 5. In order to ensure the required flow rate of the hydraulic turbocharger, an auxiliary pipeline is opened when the inlet flow rate is greater than the required flow rate on the pump side to adjust the flow to ensure the designed flow rate on the pump side.

3.3. Turbocharger performance testing results
The performance testing results of the hydraulic turbocharger are shown in Figure 11. The efficiency, head and shaft power curves of the turbine and pump side are shown to meet the design requirements. From the efficiency curve of both sides, the total efficiency of the turbocharger can be obtained.

3.4. Practical operation results
Hydraulic turbocharger has been successfully applied in the process of ammonia decarbonization and wastewater treatment system [8, 9], as well as seawater desalination process. The pressure energy recycling system based on the hydraulic turbocharger presented in this paper was applied in the seawater desalination system of Dagang Power Plant in Tianjin, China from July of 2018 and achieved stable operation.
Figure 11. Diagram of hydraulic turbocharger performance test system.

The comprehensive energy recovery efficiency reaches about 72%. Figure 12 and Figure 13 show the on-site operation diagram of the hydraulic turbocharger and the system operation data record for a certain period of time.

Figure 12. Tianjin Dagang power plantsite operation diagram.

1-Pump outlet pressure; 2-Turbine inlet pressure; 3-Pump inlet pressure; 4-Turbine outlet pressure; 5-Efficiency

Figure 13. Field operation parameters record.

The CFX simulation results show that the highest efficiency can reach 74%, and the efficiency of Dagang Power Plant in Tianjin Development Zone is about 72%. The reason of that the simulation calculation result is higher than the experimental one may lay in the mechanical loss caused by the bearing and the mechanical seal which is neglected in the numerical calculation process. However, the error between the two is very small, which shows that the design of the hydraulic turbocharger based on numerical simulation is reliable and accurate.

4. Conclusions

The hydraulic turbocharger is designed and applied in the reverse osmosis seawater desalination system to solve the problem of energy waste in the existing ones. The hydraulic turbocharger has the advantages of simple structure, low cost, low energy consumption and wide application range, and achieves stable operation, low noise and high energy recovery efficiency in the practical application. This research may provide references for the design and application of pressure energy recovery devices.
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