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Membrane feedpump optimization for efficiency

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ABSTRACT

In recent years, the size of reverse osmosis plants has increased dramatically. The layout of many plants has also changed, incorporating a center design. The advantage of this design is that the membrane feedpumps discharge into a common header, and the output of the pump is no longer linked to the flow for a specific rack of membranes. This has allowed the use of much larger pumps in a two- or three-stage configuration. The use of larger pumps improves the unit efficiency while lowering the installation costs by reducing the number of pumps. Pump manufacturers continue to strive to improve the efficiency of these pumps by modifying and upgrading the basic design. The horizontal split case design has been manufactured for more than 80 years. In the last 5 years, the efficiencies of this design have been improved substantially, thereby lowering the operating costs of the desalination plant. A paper was presented earlier outlining the role of the suction impeller and suction bay design. This paper will detail the hydraulic optimization that has been applied to maximize the efficiency.

Keywords: RO plants; Energy efficiency; Membrane feedpumps; Increased pump size; Lower installation and operating costs; Hydraulic and efficiency optimization

1. Introduction

In recent years, the size of reverse osmosis (RO) plants has increased dramatically. The layout of many plants has also changed, incorporating a center design. The advantage of this design is that membrane feed pumps discharge into a common header, and the output of the pump is no longer linked to the flow for a specific rack of membranes. This has allowed the use of much larger pumps in a two- or three-stage configuration (Fig. 1).

The use of larger pumps improves the unit efficiency while lowering the installation costs by reducing the number of pumps. Pump manufacturers continue to strive to improve the efficiency of these pumps by modifying and upgrading their basic designs. The pump choice for this market is classified as category BB3, which is a multistage, double-volute pump, with opposed impeller design. This horizontal split case between bearing design pump has been manufactured for more than 80 years. However, the efficiencies of this design have been improved substantially in recent years, thereby lowering the operating costs of the desalination plant. A paper was presented earlier [1] outlining the role of the suction impeller and suction bay design. This paper will detail the optimization that can be achieved in the cross-over from one stage to the next.

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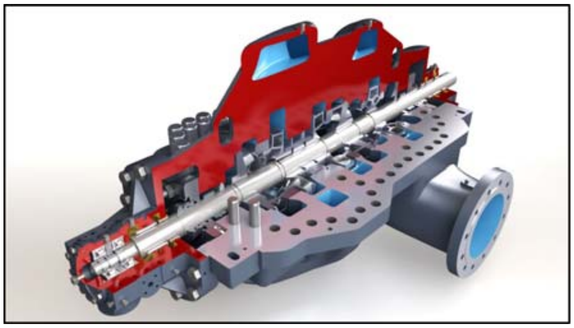

Fig. 1. A multistage volute pump for membrane feed pump applications.

In a centrifugal pump, all the energy is generated by the impeller. This rotating component produces the energy or total head by centrifugal force. The stationary component of the hydraulic system, housed in the pump casing, has been previously considered as only the means to collect the flow and guide it to a suitable discharge opening (Fig. 2).

But the two components, stationary and rotating, do not act independently. The stationary component, if not designed properly, will cause the impeller to work very inefficiently. The simplest way to view the flow coming off the impeller is to consider it a spiral flow whose circumferential component of velocity is related to the head, while the radial component is related to the flow delivery (Fig. 3).

In order for the rotating element to produce energy or head, the impeller accelerates the fluid in a tangential direction, to very high velocities. The increase in velocity is the head generated by the impeller (i.e. the head generated is from velocity head ($V^2/2g$). The tangential component of the fluid is much higher than the radial component of the flow. Therefore, the absolute fluid velocity (i.e. resultant of the flow and head vectors) exiting the impeller is also very high and must be controlled by the stationary hydraulic component.

The throat area in the volute must be sized to match the absolute velocity of the fluid coming off the impeller. Most designs follow the constant angular momentum rule. The angular momentum of the fluid at the impeller OD (D2) must match the angular momentum of the velocity in the center of the throat, D3 (Fig. 4).

Also, the angle of the cutwater tip must match the angle of the flow exiting the impeller. But this can only occur at one flow point, which is the best efficiency point on the pump curve (BEP). At higher and lower flows, the flow angle approaching the throat will be greater than or less than the angle that the cutwater was set at. This is called positive or negative incidence angles.

Fortunately, the capacity demand for this service does not vary too much over its service life, typically, 90–110% of the original design point. Therefore, the impeller will operate near or at its BEP point for the majority of its life, and the incidence angle is optimized.

The large pumps used as the membrane feed pumps in a RO facility are a double-volute design, with symmetrical volutes in each case half. The two throats will be located radially 180° apart, at equal distance from the pump centerline, and each will have half the throat area of a single volute. The velocity in the throat will be the same for either a single or double-volute designed for the same flow conditions.

Two (or double) volutes are required to balance the radial thrust produced by the impeller. The hydraulic radius of a channel opening is the area of the opening divided by its circumference. For double-volutes, this means the hydraulic radius will be approximately 70% smaller than that of a single-volute pump. Therefore, the percentage of side wall surface area in the channel will be greater in a double-volute pump than in a single-volute pump. More exposed surface area will equate to higher friction losses in the volute. Surface finish in the high-velocity areas in the throat must be controlled to limit the negative effect double-volutes have on the pump's performance. Normally, the high-velocity

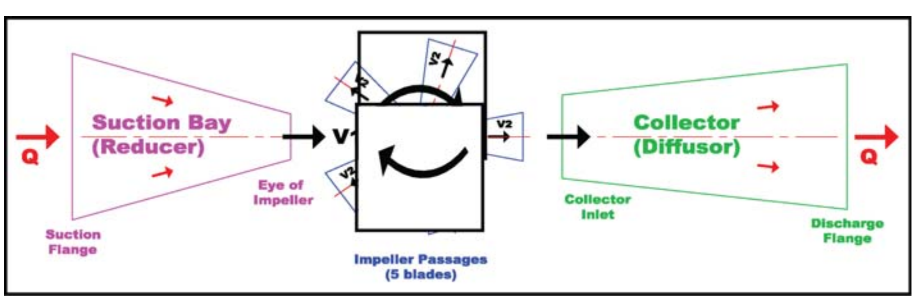

Fig. 2. The stationary and rotating components of a centrifugal pump.

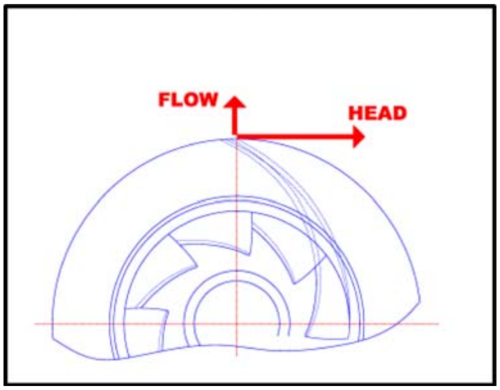

Fig. 3. The flow components of an impeller.

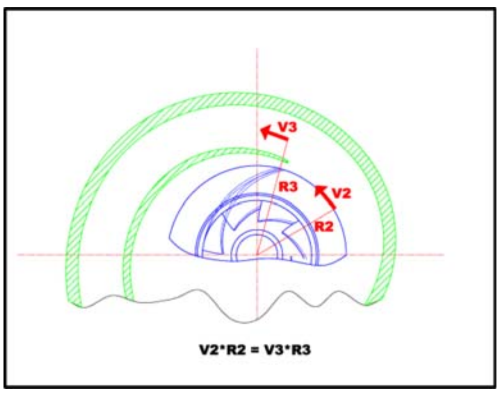

Fig. 4. Velocities coming off the impeller.

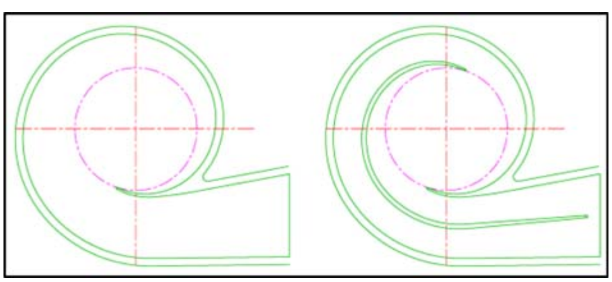

Fig. 5. Single and double-volutes.

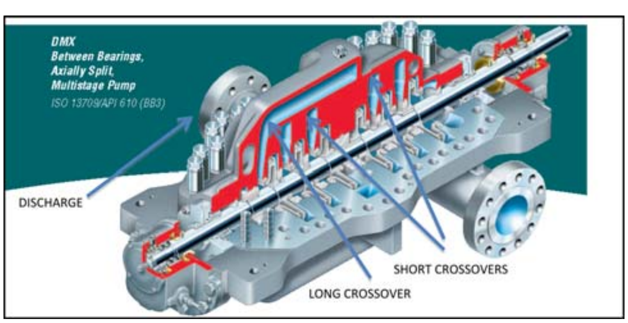

Fig. 6. The cross-over passages in a multistage volute pump.

areas in the volute and cross-over will have the surface finished improved to minimize losses due to friction (Fig. 5).

The purpose of the hydraulic channel after the throat area is to direct the flow to the next stage in a multistage pump or to the discharge opening. Multistage pumps used for RO service will have three different hydraulic channels:

- Cross-over from one stage to the next (short cross-over).
- Cross-over from the low-pressure side of the pump to the high-pressure side (long cross-over).
- Last-stage discharge channel (Fig. 6).

Guiding the flow represents the practical use of the hydraulic channel. But more important is the hydraulic design of the cross-over or discharge channel. As mentioned, the impeller is the hydraulic component that produces the head of the pump. The volutes and cross-over channels must have an optimized design to limit the loss of head when trying to direct the flow.

A cross-over is designed to diffuse, or slow down, the velocity of the fluid from the throat area to either the next-stage impeller or the discharge nozzle. The energy is converted mainly from velocity head at the throat to pressure head in the channel by using Bernoulli's equation. This is called pressure recovery, and we want to maximize this recovery (Fig. 7).

Diffusion is required because the velocity entering the next impeller or discharging from the pump cannot be as high as the velocity at the throat. As an example, the throat velocity can be 25 m/s (80 ft/s), but at the discharge nozzle, the limit should be set at 8 m/s (25 ft/s) or less.

Designing the last-stage discharge nozzle is not as challenging as designing the two cross-over channels in a multistage pump. The discharge nozzle in a multistage pump has the same features as any discharge nozzle in a volute pump, and meeting the hydraulics requirements of a good channel design is not that difficult. So this article will focus on the short cross-over and long cross-over designs.

The functions of these cross-over channels are the following:

- Convert the high-velocity fluid at the throat to acceptable velocities for the suction of the next-stage impeller.
- Change the flow pattern from mainly tangential coming off the impeller discharge to mainly radial for the next-stage impeller suction.

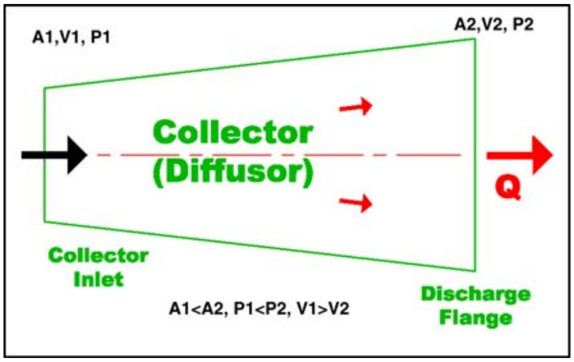

Fig. 7. Pressure recovery in a diffuser.

- Turn the fluid 180° from outward flow away from the pump to inwards flow coming into the pump.
- Provide a uniform flow into the next-stage impeller eye.

There are methods to predict the efficiency of centrifugal pumps. The most popular method comes from the study by H.H. Anderson. His prediction is based on the pump's specific speed and pump size. Fortunately in the RO market, the pump sizes are increasing and the Ns is approaching the values where the efficiency is maximized.

However, there is always scatter with test data. For a given pump size and specific speed (Ns), there is a range of achievable efficiency values. Our task is to guarantee the highest efficiency by optimizing the design and minimizing the losses. The challenge to the pump designer is to meet those predicted efficiencies with a high degree of repeatability and confidence, providing required performance guarantees.

The pump's overall efficiency is contingent on three different hydraulic factors:

- Mechanical losses: bearing, seals, and disc friction.
- Volumetric losses: internal leakage and leakage through seals.
- Hydraulic losses: turning, velocity changes, and loading.

The first two components, mechanical and volumetric losses, can be controlled by the design of the rotating components, mainly the impeller and wear components. These losses are well documented and can be calculated easily. Hydraulic losses can be controlled by the impeller design and the hydraulic channel design of the cross-over. Channel designs are usually overlooked in the design of a multistage volute pump.

The design of the cross-over tends to be compromised by manufacturing limitations and mechanical design. Considerations must be given to the casing that houses the cross-over:

- Pressure boundary: the casing must be designed to contain the pressure that is generated.
- Casting methods: this type of pump with its multiple passage-ways can be difficult to cast.
- Finishing and access to passages: internal passages need to have access for cleaning and improving the surface finish.
- Cost: the weight and size of the casing must be considered to make the pump marketable.

The hydraulic needs of any cross-over are the following:

- Diffusion utilizing linear area changes.
- Low velocity to minimize friction losses.
- Gentle turns to minimize turning losses.
- Acceleration into the next impeller matching its pre-rotation geometry.

There is usually a compromise between trying to obtain mechanical integrity in a low-cost casing and maximizing the pump's efficiency.

Fig. 8 depicts three different conventional cross-over designs. The rule is that the tighter the cross-over, the greater the losses. This is shown on the curve for the three different designs. The tightest turn will produce the most compact casing casting at the lowest cost while sacrificing the efficiency of the pump. The design with long straight lengths of steady flow will produce the most efficient pump while having to overdesign the case. This will increase the casting cost by requiring more material due to the larger geometry with thicker walls to maintain the same pressure rating. Also, there are designs that require the cross-over passage to be a separate casting or fabrication and welded or flanged to the casing to complete the cross-over (Fig. 9).

It has been shown experimentally that conventional cross-over designs do not provide a uniform flow pattern in the passage-way and the correct flow pattern to the next-stage impeller. Fig. 10 shows the flow pattern in a typical conventional cross-over design.

From this figure, you will see that the flow tends to adhere to the outside wall, which will set up secondary flow patterns and non-uniform flow to the next-stage impeller. Both of these undesirable conditions will increase the energy losses in the cross-over and reduce the overall efficiency of the pump.

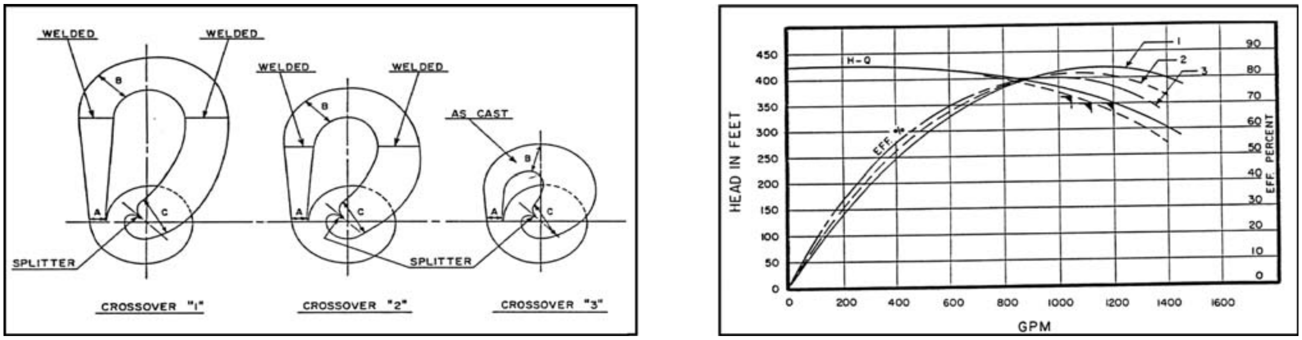

Fig. 8. The relationship between the shapes of the long cross-over and the slope and efficiency of the performance curve [2].

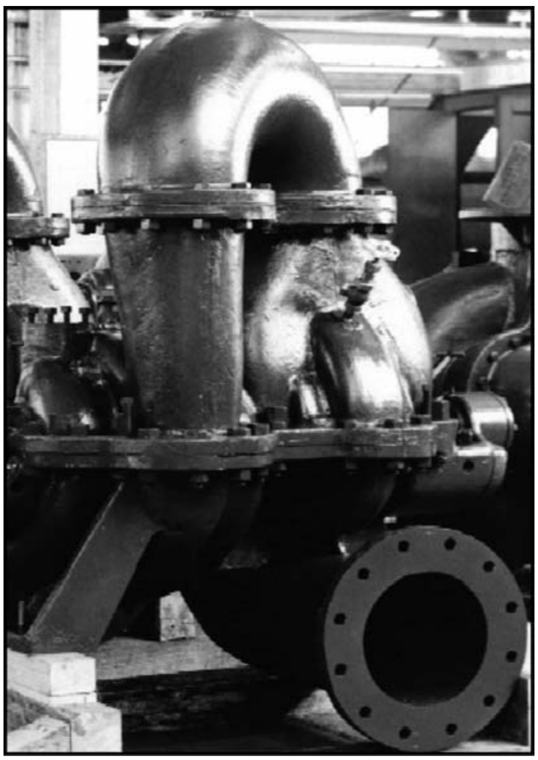

Fig. 9. A BB3 pump with a flanged elbow in the cross-over.

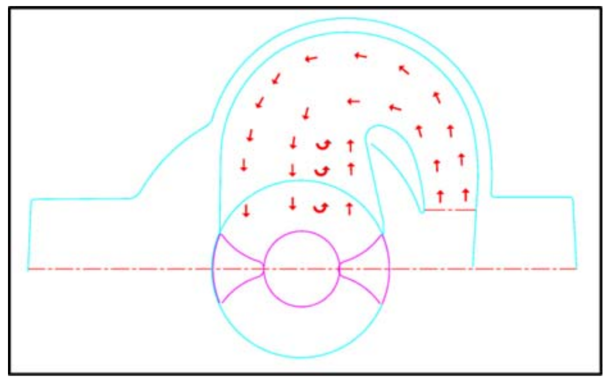

Fig. 10. Flow pattern in a conventional cross-over design [3].

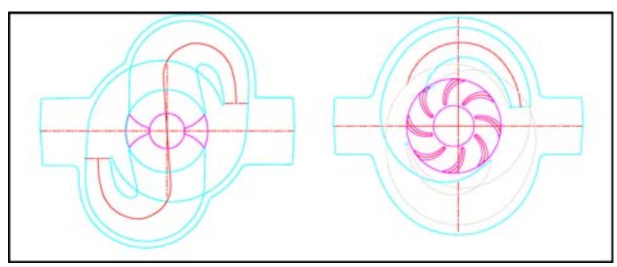

Fig. 11. A conventional cross-over (left) and continuous cross-over (right).

Fig. 11 (left) shows the conventional cross-over design with the following attributes:

- Short diffusion length: the diffusion process ends prior to the turn of the passage-way.
- Sharp radius turn: the flow turns 180° while utilizing only one quadrant of the casing cross-section.
- Development of the hydraulic sections in such a short space leads to limited section geometrical control
- Nonlinear area progressions as well as nonlinear aspect ratios lead to abrupt changes in these hydraulic parameters.
- Lack of control of the flow exiting the hydraulic passage leads to the flow angles not matching those of the impeller.

Fig. 11 (right) is the continuous cross-over design that has the following attributes:

- Continuous diffusion over most of the length of the passage—the diffusion process starts at the throat area in the casing and continues for approximately

80% of the channel length, where the fluid is then accelerated along the remaining length into the next impeller eye.
- The cross-over is one smooth long radius turn, utilizing more than two quadrants of the casing cross-section.
- A longer hydraulic path allows for greater control of the geometry of each hydraulic section.
- Linear area progression and aspect ratios lead to smooth transitions between sections.
- The fluid is controlled and directed into the next impeller by utilizing guide vanes at the end of the cross-over. This allows the designer to easily match the flow coming off the cross-over with the blade angles of the next-stage impeller.

The continuous cross-over design has the following advantages over the conventional cross-over design.

- Hydraulic shapes are optimized to minimize changes in velocity and losses.
- The cross-over delivers uniform flow to the following stage.
- Mechanical design is optimized by maximizing distribution of material to achieve pressure integrity, easily castable at a competitive cost.

Traditional designs for volutes and cross-over in an axially split pump were drawn in two-dimensional format. However, current technology employs solid modeling and utilizes computational fluid dynamics (CFD) design methods. This three-dimensional approach to hydraulic design can identify geometrical issues, which may potentially affect performance. Looking at Fig. 12 which is a solid model of a traditional long cross-over design, one can see potential turning and dumping losses with no flow control or pre-rotation into the next-stage impeller.

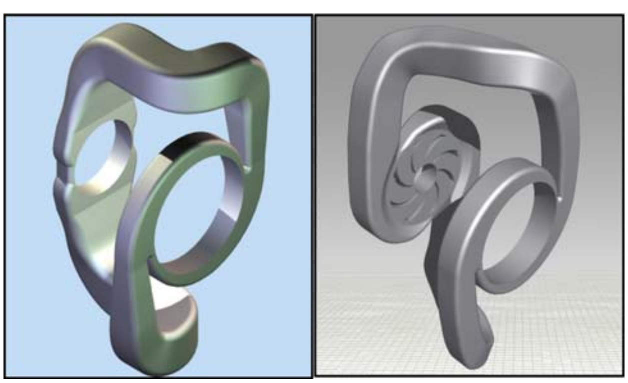

Fig. 12. CFD designs of a conventional cross-over (left) and continuous cross-over (right).

Designing in three dimensions and using CFD have the following advantages:

- Control of sections in all three planes: This will lead to the optimized shape of each section throughout the length of the cross-over. Adjustment to aspect ratio and area progression based on length and shape of the center stream line can be made easily. Then, the desired area progression and aspect ratio throughout the channel will be obtained, which create high-efficiency pumps.
- Optimizing radii of turns: High-velocity changes in the short radius turns equate to high loss coefficients.
- Visualization of mid-streamline curvature: Smoother curves streamline the fluid path in the continuous cross-over design. Changes in velocity are gradual, which equates to lower losses.
- Pre-rotation to the next-stage impeller is controlled by the guide vanes.

In addition to the computational fluid dynamic analysis, the solid model is used in the mechanical design of the casing. Finite element analysis (FEA) is performed to verify the integrity of the casing by:

- Analyzing the stresses throughout the casing, to assure that they do not exceed allowable values.
- Looking at the deflection of the casing at certain areas to assure the pump will operate reliably throughout its service life.

Also, the solid model of the casing is used to do pour simulations at the foundry to verify the integrity of the casting. Design changes can be made prior to pouring the casting, to eliminate any areas with high risk of casting defects.

The method of design and manufacturing described in this article has allowed the state of the art to be dramatically improved over the last five years. With the combination of CFD hydraulic design, FEA mechanical design, and pour simulation at the cast level, we can produce a pump that has:

- The highest efficiency obtainable and predicted efficiencies met with a high degree of repeatability.
- A casing with a mechanical design optimized to minimize material and cost while achieving pressure integrity, and reliability in service.
- Few casting defects and easily castable.
- High market capabilities.

Below is an example of a typical application for a BB3 pump in RO service with the financial benefit of the continuous cross-over design:

Pump = 12 × 22 DMX-2 Stage.
Fluid = Sea Water (SG = 1.03).
Flow = 2200 m^3/h, head = 600 m, speed = 2990 rpm, Ns = 1675.
Efficiency of a conventional cross-over = 85%, power = 4350 Kw.
Efficiency of a continuous cross-over = 88%, power = 4200 Kw.
This is an energy savings of 150 Kw.

If the cost of electricity is $0.07/kWh and the pump is typically run for 8000 h/year, the cost savings per year are as follows:

$$150\ \text{kW} \times \$0.07/\text{kWh} \times 8000\ \text{h/year} = \$84,000/\text{year}$$

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