Design Status of High-Pressure Water Injection Pumps
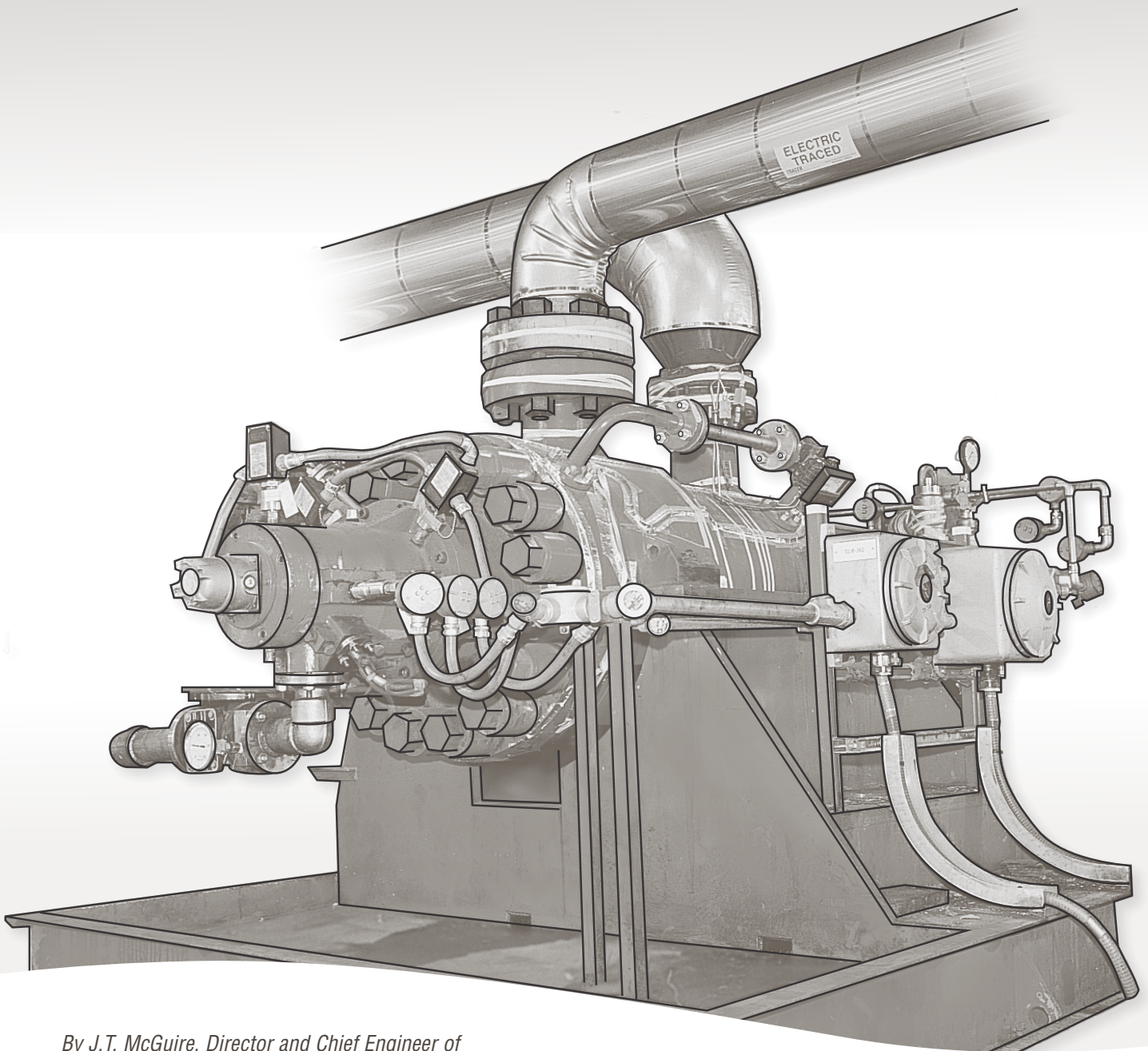

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Introduction

Water injection (aka *waterflooding*), a form of secondary enhanced oil recovery (EOR), was first applied by oil producers in Pennsylvania in the early 1920s. In addition to restoring depleted pressure within the reservoir, it also helped transport the remaining oil to the production wells, where it could be recovered. By the early 1970s, injection pump discharge pressures had already risen to 355 bar (5200 psig). Pumps in service today are rated for 520 bar (7500 psig) discharge pressure with drivers rated for 11 400 kW (15 300 hp) [Figure 1], while others currently in production are rated for 620 bar (9000 psig) discharge pressure and absorb 11 500 kW (15 400 hp) at rated flow. Drawing from projected hole bottom pressure requirements, pumps producing 1030 bar (15 000 psig) injection pressure will be in the field by 2020. To date, each advance in generating higher injection pressure has presented significant engineering and manufacturing challenges and doubtless will continue to do so. The objective of this paper is to review the status of contemporary water injection pump design in order to provide a baseline for the design of the generations to come.

While many types of pumps have been used to extend the life of declining oilfields both onshore and offshore, today most are process barrel-type pumps. Designed and engineered to the latest issue of ISO 13709/API 610 standard, API Type BB5 barrel pumps are among the most technologically advanced industrial centrifugal pumps ever built (excluding aerospace), with envisioned performance soon likely to reach 3600 m³/h (16 000 gpm), heads to 6000 m (20 000 ft) and power to 27 000 kW (36 000 hp).

This review of current high-pressure water injection pump design status is set out in five sections:

1. Rotor

2. Hydraulics

3. Stator

4. Pressure boundary

5. Materials

Figure 1: *Water injection pump, offshore, CGT drive*

Rotor

Classes

There are two fundamental classes of rotor design: *slender* or *large shaft*. The former has relatively low mechanical stiffness; the latter has significantly higher mechanical stiffness. This distinction and its influence on attainable shaft and rotor runout along with assembled rotor balance are addressed in the Shaft Flexibility Factor, L^4/D^2, stated in Section 9.2 of API 610, 11th Edition (ISO 13709:E [2009]). The choice of rotor class affects the hydraulics for a given application and the potential mean time between having to renew running clearances (aka *MTBR*). Therefore, in most cases, the selection of rotor class is the first and most important decision the designer has to make.

Classification criteria

Two criteria are used to classify rotors as *slender* or *large shaft*. The first is static deflection, i.e., rotor centered in the first-stage impeller front hub and balance drum or sleeve running clearances [Figure 2], relative to the minimum new running clearance at rotor mid-span. A *slender shaft* rotor's static deflection is typically greater than 0.50 of the minimum new running clearance at rotor mid-span. This results in the rotor resting on its running clearances when the pump is stationary, thus precluding what is termed a *rub-free* build. In comparison, a *large shaft* rotor's static deflection when centered in its running clearances as shown in Figure 2 is typically 0.25–0.30 of the minimum new running clearance at rotor mid-span.
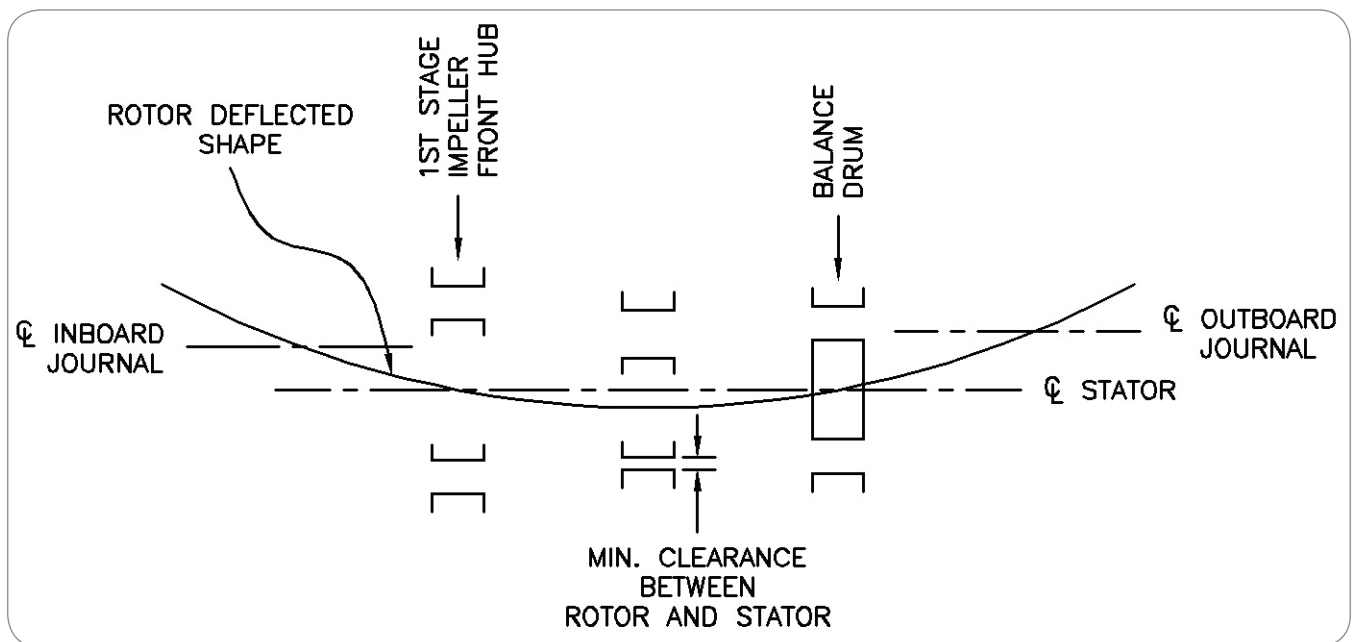

Figure 2: *Rotor radial centering diagram*

The second distinction is dynamic and based upon a chart originally developed by Duncan and Hood[1] that classified rotors by a factor, K = (WL³/D⁴) 0.5, where W is rotor weight, L is bearing span and D is shaft diameter at the impellers, all in consistent units. This factor is mathematically related to the rotor's first dry bending critical speed. Duncan and Hood's original chart [Figure 3] shows the *slender* and *large shaft* rotor classes, but also includes rotor classes *too slender* and *dry running*. Operating experience, both intentional and unintentional, has verified the *dry running, large shaft* and *too slender* characteristics.
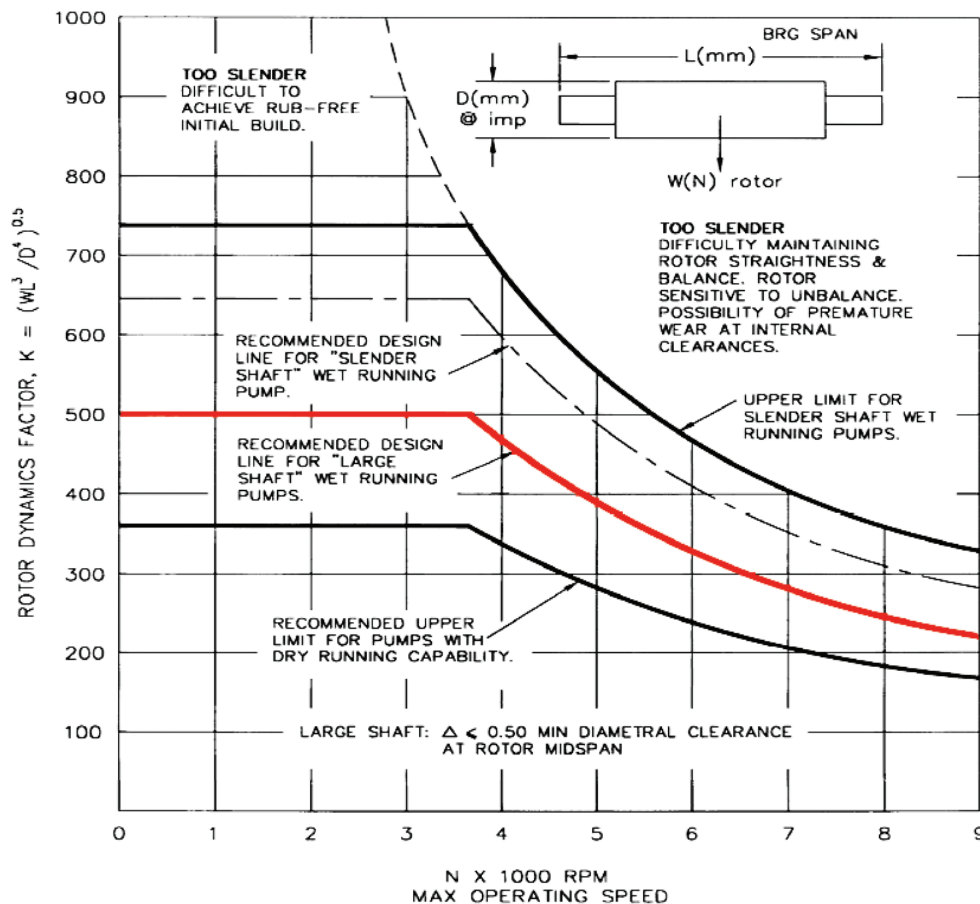

Figure 3: *Rotor dynamics factor, K, vs. running speed after Duncan and Hood* [1]

(1) A. B. Duncan and J. F. Hood, *The Application of Recent Pump Developments to the Needs of the Offshore Oil Industry*, Proc. of the Conference on Pumps and Compressors for Offshore Oil and Gas, London, UK, June 29-July 1, 1976, pp 7-24.

Rotor dynamics

In rotor dynamic terms, the virtue of *large shaft* and *run-dry* rotors is that the hydraulic stiffness and damping produced by the Lomakin Effect[2] has much less influence on the dynamic behavior of those rotors than on a *slender shaft* rotor. The net result is greater tolerance of the gradual increase in running clearances during operation, thus longer mean time between repair (MTBR).

The initial design of *large shaft* rotors usually has an operating speed above their first dry bending critical speed and no higher than 0.85 of their second dry bending critical speed. Rotor dynamics analyses are run as prescribed in Annex I of API 610, 11th Edition/ISO 13709:E (2009), and the design must satisfy the separation and damping requirements of Appendix I.

Depending upon the starting frequency and the severity of the application, the running clearances of *slender shaft* rotors must be renewed every 5000 to 50 000 hours. In comparison, those with *large shaft* rotors typically log 100 000 hours of run time between running clearance renewal. Run times of 150 000 hours before running clearance renewal are now deemed generally feasible. It should be further noted that run times over 200 000 hours have been demonstrated in a number of installations, albeit at lower pressure rise and power than now necessary.

Shaft size affects hydraulic design. The generally accepted rule is a shaft larger than that needed for acceptable torsional stress has a deleterious effect on NPSHR and efficiency, hence power. There is some truth to that, but with the design tools available today it is practicable to reduce those effects so they effectively fall within the uncertainty of test efficiency.

A second effect that becomes apparent as pump power rises to 15 000 kW (20 000 hp) is a design conflict between the shaft diameter needed for acceptable torsional stress and the maximum allowable shaft diameter as a fraction of impeller (OD/D_2). Resolution of this design conflict is central to achieving acceptable hydraulic performance.

Impeller arrangement and mechanical design

Impeller arrangement on the rotor depends upon pump pressure rise, or differential pressure (ΔP). In general, tandem or in-line impellers (with a balance drum to compensate for impeller axial thrust) are used for ΔP up to 400 bar (6000 psig). For higher ΔP services, the variation in rotor residual axial thrust reaches values too high for practical thrust bearings, particularly as pump speed rises. To accommodate these conditions, the impellers are opposed, with a center sleeve and balance sleeve to compensate for axial thrust. Figure 4 shows a section drawing of an opposed impeller rotor in a radially split diffuser stator.
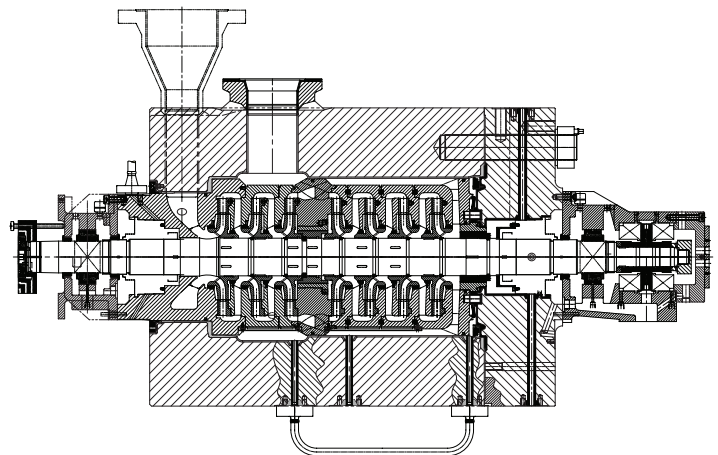

Figure 4: Section drawing of a seven-stage opposed impeller diffuser pump

(2) The Lomakin Effect is a product of hydrostatic (pressure) and hydrodynamic (velocity and viscosity) action as liquid passes through a pump's internal running clearances under the action of a differential pressure. As such, the magnitude of the Lomakin Effect depends on the pressure drop across the running clearances, the surface speed at them, their geometry and the liquid viscosity. Given these dependencies, there is very little Lomakin Effect until the pump is running at say 75–80 percent of rated speed. [McGuire & Karassik, Centrifugal Pumps, 2nd Edition, Chapman and Hall, 1996.]

With rated speed of high-pressure water injection pumps now at 8000 rpm, power per stage at 1500 kW (2000 hp), and a pressure rise to 74 bar (1070 psig) per stage, it is necessary to conduct a dynamics analysis of each design of impeller used on a given rotor to verify that:

a) Stator vane passing frequencies do not excite a mode of the impeller shrouds' natural frequencies [Figure 5]

b) Fatigue life at high stress points in the impellers is acceptable [Figure 6]

There have been instances of impeller failure that reinforce the need for impeller dynamic analysis.

Running clearances

The running clearance surfaces of the impellers, balance drum and center sleeves are hard coated to improve resistance to erosion and galling in the event of incidental contact during operation. Direct laser deposited tungsten carbide is the usual hard coating material.

Rotor assembly

The means of installing and removing the rotor's mounted components — impellers, sleeves, balance drum, thrust collar and coupling hub — are interference fits, each located on a different shaft diameter. Components or assemblies that are replaced during field maintenance, e.g., shaft seals and bearings, first require removal of the pump's coupling hub and thrust collar, both of which are installed and removed using oil injection shrink fits. Shaft seals are typically cartridge type, mounted with a clearance fit on two diameters to minimize the length of close-fit engagement during installation and removal.
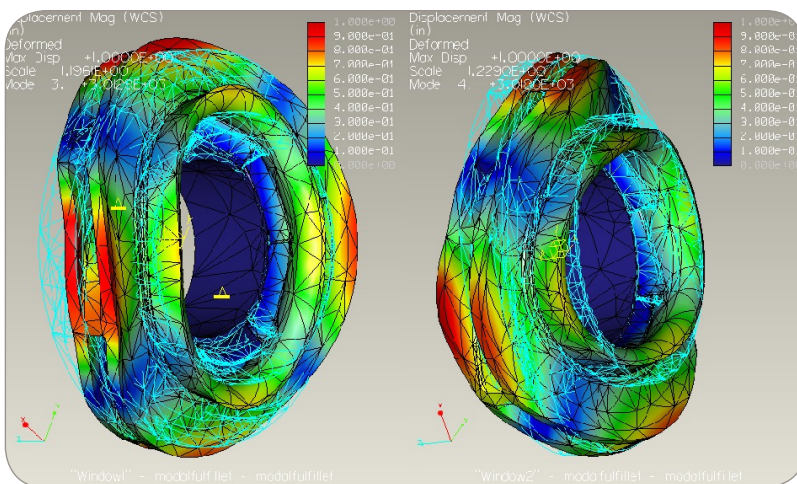

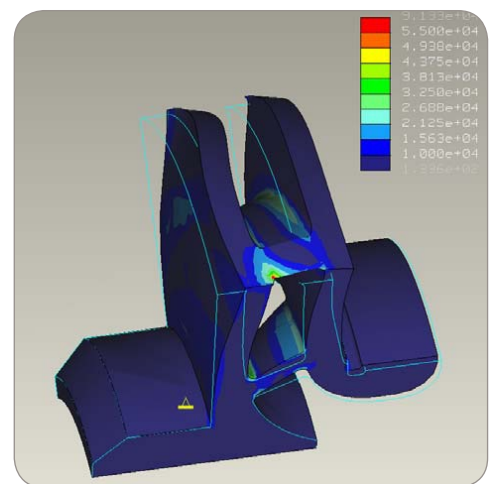*Figure 5:* Impeller dynamics analysis:
0 circle – 2 diameter (0.2) mode

Figure 6: Stress distribution in impeller discharge vane-shroud junction

Hydraulics

The typical seawater injection system today has a minimum of the following equipment/systems upstream of the injection pump:

- **Seawater lift pump(s)** – raise water from the ocean for treatment before injection

- **Deaerator** – lowers the O_2 content of the injection water

- **Water treatment (O_2, S)** – further removal of O_2 and removal of sulfur

Selection criteria for the booster pump must include developing sufficient head to have NPSHA at the injection pump above incipient NPSH ($NPSH_i$) over the pump's operating flow range.

Specific speed (Ns), number of stages (n) and rotative speed (N) are intrinsic to rotor design for bending stiffness and to develop the required pressure rise at rated flow. Given the interaction between rotor and hydraulic design, the development of the final design is necessarily iterative.

Inlet peripheral velocity, U_1, is now typically ≥ 60 m/s (200 ft/s). Therefore, the first-stage impeller has to be designed for incipient NPSH below NPSHA throughout the pump's operating flow range in order to avoid cavitation erosion. Achieving this often requires an unconventional inlet design, potentially affecting both the suction guide in the casing and the inlet geometry of the first-stage impeller. The former provides close to uniform flow distribution into the first-stage impeller; the latter minimizes the pressure reduction on the underside of the inlet region of the vanes. Flow analysis by computational fluid dynamics (CFD) has produced significant gains in this aspect of hydraulic design and in diffusion of the high-velocity flow leaving the impeller.

The selection of double-suction, first-stage impellers to lower pump NPSHR is generally counter-productive in high-pressure water injection pumps for two reasons:

- The number of stages is effectively reduced to $n+(0.7)^2 = n+0.5$, where n is the number of single-suction impellers. This is because the feasible diameter of a double-suction first stage in a type BB5 pump is nominally 0.7 of the series stages, a consequence of the crossovers needed to conduct water to the second-stage impeller.

- The inner side of the first-stage impeller has two obstructions (i.e., crossovers) in its inlet passage when the pump has a radially split stator. There are four obstructions (i.e., the crossovers plus the split joint flange) when the inner casing is axially split.

Generated head per stage is currently on the order of 725 m (2400 ft) and has proven viable in operation in the Gulf of Mexico. A design for 760 m (2500 ft) of head per stage is in production. Interestingly, a pump with even higher operating ratings — a four-stage, 11 000 rpm pump developing 860 m (2800 ft) of head per stage — was installed on a platform in the North Sea circa 1980. Data on the operating history of this pump, however, are meager.

Stator

Two stator designs are employed in the inner casing: diffuser [Figure 7] and twin volute [Figure 8]. In almost all cases, diffuser stators are radially split. The complexity of producing zero-leakage axially split diffusers to allow installation of assembled rotors (as is done in centrifugal compressors) has been resolutely determined impracticable in terms of cost vis-à-vis benefit analyses.
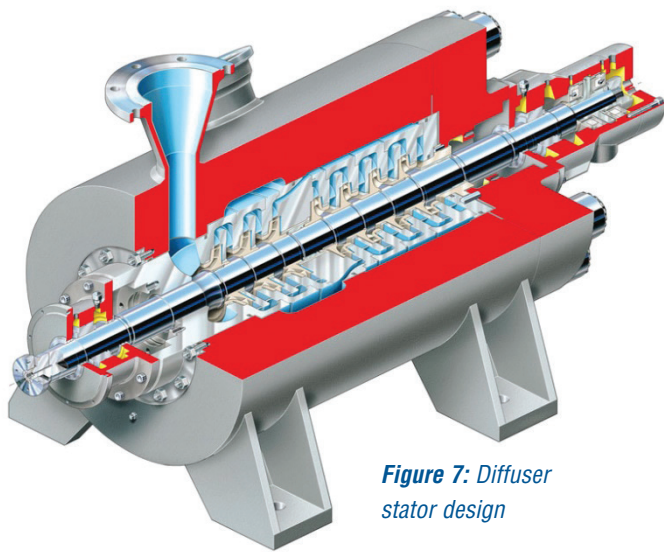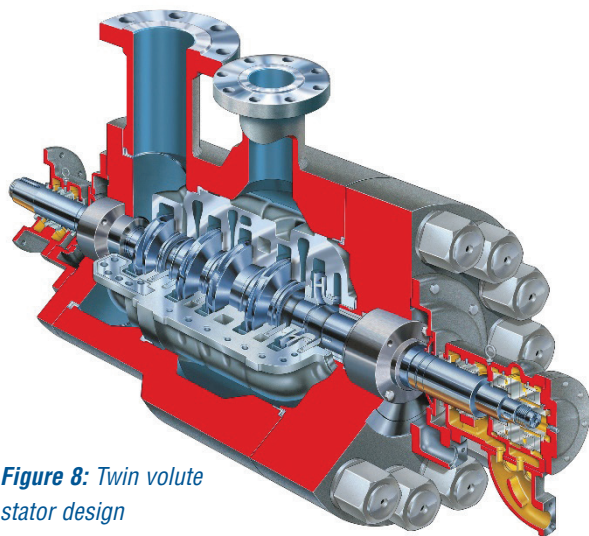

Figure 7: Diffuser stator design

Figure 8: Twin volute stator design

Double volute stators are generally axially split, a feature found in API type BB3 pumps. These pumps were the first used in high-pressure injection service (often in series) and influenced subsequent water injection pump designs.

The first aspect of stator design to be considered is the accurate reproduction of the stator's hydraulic geometry and passage surface finish. Diffuser stators typically have diffuser passages milled into the stage piece casting when cantilever vanes are acceptable. This results in highly accurate reproductions of geometry and excellent surface finish. When cantilever vanes are not acceptable, the diffuser is produced as a separate precision casting, machined as necessary and finally attached to the stage piece on an interference centering fit. Pins are used to accommodate the reaction torque produced between the rotor and stator. Similar geometry reproduction and surface finish are feasible in large axially split, double volute stators. Achieving this, however, becomes more difficult as the stator decreases in size, thereby limiting access to the long, small cross-section passages inherent in that design.

A second critical issue to assess is the effect of stator diameter on the pressure boundary design as the rated flow and discharge pressure rise. This consideration puts axially split, double volute stators at a disadvantage because their outside diameters are larger than equivalent diffuser stators, thus increasing the diameter of the casing and casing cover and therefore, the weight of the pump. With maximum allowable working pressure (MAWP) already at 810 bar (11 750 psig), every design feature contributing to lower gross weight becomes very important.

Pressure boundary

The pressure boundary includes the pump's shaft seals. When the casing is rated for a single MAWP that is based on discharge pressure, the shaft seals must be capable of containing that pressure in a static state. The pressure rise across high-pressure water injection pumps is already so high that rating the entire casing for one pressure level is impractical. A number of oil companies have come to this conclusion and now allow the pressure boundary to be rated for two pressures: one MAWP based on maximum discharge pressure; the second lower MAWP based on the highest operating pressure in the low-pressure regions of the casing.

MAWP of the low-pressure regions is unlikely to be greater than 140 bar (2000 psig) and is therefore designed to ASME Section VIII, Division 1 or 2.

When MAWP of the high-pressure region is ≥ 700 bar (10 000 psig), the required design code is ASME Section VIII, Division 3. The analysis requirements of Division 3 are intended to model the behavior of the actual material at a local level, thus finite element analysis (FEA) is mandatory. Given the better knowledge of the local stress, the hydrostatic test pressure ratio is reduced to 1.25 times MAWP.

Materials

Most high-pressure water injection pumps built today are for offshore installations. As such, materials of construction generally are Material Class D2, 25Cr duplex stainless steel, as stated in Annex H of API 610, 11th Edition/ISO 13709:E(2009), with the following comments (see Table 1):

Table 1: *Comments regarding Materials Class D2, per Annex H of API 610, 11th Edition/ISO 13709:E(2009)*

MATERIALS	
Shaft	Forged and rough machined bar of A182, Gr 55, UNS S32750, has proven more stable than hot rolled bar, thus allowing manufacture of shafts whose runout stays within allowable limits.
Impellers	Precision casting, A890, Gr 5A, radiographic quality. EDM was used by owner circa 1980 to produce replacement impellers from forged Ferralium 255 for waterflood injection pumps. EDM has not been used for production pump impeller manufacture to date, although it is now common practice for centrifugal compressor impellers. Hubs are hard coated with DLD of WC (typically 60WC/40Ni).
Balance Drum	Forged A182, Gr 55, UNS 32750, OD hard coated as are impeller hubs
Diffusers	As impellers if separate
Stage Pieces Inner Casing (twin volute)	Sand cast, ASTM A890, Gr. 5A
Wear Rings	Bar stock, UNS 32750, hard coated with DLD WC (typically 60WC/40Ni)
Casing	Forging. Risk of sigma phase formation limits allowable production thickness of super duplex castings or forgings to a conservative 200 mm (8 in). As necessary, wall thickness for high-pressure water injection pumps is usually > 200 mm (8 in), forged carbon steel with all wetted surfaces weld overlaid with Inconel 625 are typically specified.

Conclusions

Much has been achieved in water injection pump design since the early 1970s. But there are more challenges to meet and pump hydraulics barriers to break. With the design tools available today, an eye to the practical and careful attention to detail, the challenges now ahead of us can be confidently confronted.

About the author

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James T. McGuire is Director and Chief Engineer of Custom Engineered Pumps at Flowserve Corporation. He has more than 51 years of experience in the pump industry, starting with Worthington in 1965. He has dedicated a considerable portion of his career to advancing the design of high-energy pumps used in oil production and refining as well as electric power generation.

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