SELECTION OF LARGE WATER TRANSPORT PUMPS AND FIELD EXPERIENCES

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ABSTRACT

This paper covers the selection criteria with respect to suction behavior and optimum efficiency of large water transport pumps to meet various system requirements. Simplicity of pump design and operating behavior of pumps with respect to low maintenance cost and good rotordynamics are also addressed.

INTRODUCTION

In view of the worldwide increase in energy costs, more attention must be given to the economics of pumping installations as a whole. The experience gathered with design of storage pumps can be used as the basis for the design of water transport pumps. The availability, ease of servicing, and life of the individual components, in relation to the hydraulics, determine these engineering plant system characteristics. Examples are given from recently commissioned pumping systems.

The criteria for selection will be shown on large water transporting systems in the Middle East and in South America. The pumping stations in large water transport systems are equipped either with parallel or in-series pumps, or a combination of both depending on the geodetic profile. It is important to achieve lowest energy cost, simplicity of operation, and operational flexibility.

Consequently, the main objective in the design of water transport pumps is a compact design coupled with ease of maintenance. The criteria for the selection of pump types for transporting water are:

- Hydraulic aspects given by:

  • Flow velocity and, hence, limitations on account of cavitation, corrosion, and abrasion

  • Efficiency, with possible penalties for loss of energy

  • Specific speed, hence, pump geometry and lifetime of parts sustaining wear

- Engineering aspects given by:

  • Test pressure

  • Rotordynamic features

- Plant system aspects given by:

  • Intake system

  • Driving types

  • Kind of liquid

  • Transient condition

Water transport pumps are selected for the supply of drinking water to cities located at high elevations with respect to the water sources, and for irrigation of plantations located far away from the water sources, e.g., in Columbia, Mexico, Venezuela, and the Middle East. In special cases, the same type of transport pumps is used to convey seawater over a distance of several hundred kilometers to injection points in the oil and gas fields to recover crude oil.

SYSTEM REQUIREMENTS

Pipe systems have to be analyzed for their initial and final flowrates and the storage capacities of the intermediate reservoirs. The plant design should offer flexibility, in that the capacities can be increased at any time without changing the pump concept. In new plants, a method of keeping the flexibility is to arrange the pumps in a pumping station in a symmetrical arrangement. In a first phase, one transport pipeline would be sufficient to cope with the flow capacity, but provisions are made to install two parallel main pipes. For each pipeline, a number of pumps plus one standby pump would be installed. The distribution of pump flows will be

done such that the minimum flowrate can be pumped in the optimal pumping range between 80 percent and 110 percent of the pump best efficiency. A good combination might be three pumps in operation and one standby pump.

The general approach for the layout of a pumping station is to collect the following technical data:

● Design flowrate, variation of flowrates (during the life of the installation)

● Properties of the pump fluid

● Philosophy for control and protection of the complete system

The definition of the station requirements is important, which should fulfill the operating conditions. The information for meeting the requirements is normally based on the layout of the system hydraulics.

HYDRAULIC LAYOUT

For the hydraulic layout of the station, it is necessary to know the properties of the fluid and the system friction curves as well as the geodetic profile along the pipeline. For different phases of the project, the system throttle curves (discharge pressure minus suction pressure is equal to the differential pressure) will be plotted against the geodetic pipeline profile. In addition to the system throttle curve, there are further main issues to be considered.

By appropriately splitting the flows or heads into several pump units, optimal combinations with reasonable driver arrangements and control should be achieved with very high overall pumping station efficiency for the whole operating range. The same splitting and pump unit sizes should be applied in any additional pumping station to reduce the spare part requirements and maintenance cost.

Open or Closed Drinking Water Transport Systems

From time to time, desalinated seawater has to be transported several hundred kilometers to the consumers. Where the distance is very large, transportation of the water can no longer be accomplished by one pumping station alone, since the frictional losses are too big. The pipelines would have to be designed to operate at higher pressures, and the pipeline wall thickness would have to be increased. Costs of the pipelines would rise with the increase in wall thickness. The extra costs for additional pumps and motors are, however, hardly significant compared with the potentially higher pipeline costs. Hence, the required total delivery head is distributed over several pumping stations. Distribution is arranged to match the topographical conditions of the terrain.

A difference is drawn between the open system, in which the pumps of the head station deliver into a reservoir tank that is under atmospheric pressure, via a pipeline system, and the closed system, where the pumps of the first station deliver directly, over a long distance, into the pumps of the next station, thereby providing the suction pressure for the following stations.

Open System

The advantages of the open system are:

● The reservoirs act as equalization tanks, thereby providing a suitable arrangement where transient operating conditions can occur.

● When the terrain is hilly, the reservoir can be sited at a greater geodetic height than the pumping station. Hence, a higher value of pressure is available at the impeller inlet, so that the pumps can be operated at a greater speed of rotation. Smaller pumps can therefore be utilized, resulting in less space requirement for the pumping station.

● Variable speed motors are not required, provided that the storage capacity of the reservoir is sufficient.

● The expenditure required to prevent water hammer can be lower.

● The pipelines can be designed for lower operating pressures.

● It is possible to tap off from the pipeline for consumers located in the vicinity of the reservoir and extract any required amount of water without disturbance of the operating behavior of the downstream pumps.

Closed System

The closed system offers the following advantages:

● A reservoir can be dispensed with, so that the civil engineering costs are lower.

● In level terrain where a reservoir would provide only low geodetic pressure at the impeller inlet, the upstream pumping station develops the inlet pressure. This inlet pressure can be set sufficiently high to allow the utilization of smaller and moderately priced pumps functioning at a higher speed of rotation.

Control of the installation, i.e., regulation of the speed of rotation, requires considerably higher expenditure. Measures to avoid nonpermissible values of water hammer can also involve significant expenditure.

The pipelines have to be designed to withstand higher pressures than for an open system, thereby increasing the costs of the installation. It is often necessary to provide speed controlled drives in order to increase the operating flexibility of such a system. The topography of the terrain has to be taken into account by the plant design engineer when deciding whether an open or closed system is to be used.

Pressure Surges in Pipeline Systems

In pipeline systems, any change in the operating state leads to dynamic pressure changes, which must be taken into account in the planning and operation of installations. Surge and water hammer are capable of causing serious damage.

Water conveyance and supply systems are often extremely varied in nature, so that it is impossible to find by approximation procedures a solution that will reliably preclude overstressing of components in any system.

In steady operation, the flow velocity is constant in time and place. By contrast in transient flow, the velocity varies in time and place. Transient flows occur at every change from an existing steady operating state to a new one. Pressure surges result from transient flows. Consequently the following operations on pumping installations are generators of surging problems:

● Starting a pump

● Stopping a pump

● Switching over pumps

● Altering the valve setting

● Altering the speed

● Power failure on one or more pumps

● Inexpert operation

With the exceptions of power failure and inexpert operation, all of them are deliberate actions that can be performed properly. On the other hand, power failures are unintentional and generally constitute the most extreme case. Additional safety devices must be provided against power failure in particular.

Basically any system may be at risk, though low-pressure installations are more endangered than high-pressure ones, because the maximum pressure change given by the Joukowsky equation is independent of the system pressure.

The purpose of every pressure surge investigation is to fix the size of a protective device so that no limits are exceeded anywhere in the plant, even in the most extreme transient case. The limiting factors are as follows:

- Minimum pipeline pressure

- Maximum pipeline pressure

- Maximum reverse speed

- Pipeline profile

The pressure surge investigation often reveals that a better line profile will allow much smaller protective devices. However, this advantage can be exploited only if the transients have already been examined at the project planning stage.

Optimal Arrangement of the Pumps Within the Pumping Station

The specifications of a proposed pumping station can be often met by a large number of different pump configurations. Each selection has an influence on the civil engineering aspects of the station (e.g., space requirement and depth of excavation necessary) as well as on the operating characteristics of the electromechanical equipment (e.g., speeds of rotation). The most economic layout of the pumping station is evaluated from the civil engineering costs and the total costs of the electromechanical equipment. Naturally the inlet structures have also to be included.

The arrangement of the pumps within the pumping station is mainly determined by the following influencing factors:

- In this paper only the horizontal pump arrangement is covered

- Pipeline layout

Horizontally split, single, or multistage, double-flow pumps are generally employed in water works because dismantling of them is easy. Suction and discharge connections are located in the lower part of the casing. These connections may be directed tangentially downward. This provides the advantage that pipelines do not impede access to the pump, but cause increased civil engineering costs. A layout incorporating upstream booster pumps can be appropriate, particularly for horizontal pumps. When a booster pump is fitted upstream, it is possible to operate the main pump at a higher speed of rotation. This is a means of decreasing the price of the pump. However additional drive motors, as well as more valves, are then required, and the space occupied is greater. In some cases the suction and discharge connection to the pump are so-called inline underneath the horizontally split main pump flange. With this pump arrangement, station piping for a parallel operating pump has to optimize with optimal bends with baffles (splitter vanes) to straighten the flow and taper pieces to accelerate the flow toward the suction side and to match the suction nozzle. On the discharge nozzle, the high velocity has to smoothly deaccelerate to the main pipe header. Double-flow two-stage pumps, called 2d, and double-flow three-stage pumps, called 3d, require breeches pipes on the inlet side, which have the task of distributing the water into the two pump suction chambers. The approach to the pump suction could be executed in many different ways (Figure 1).

The design of the breeches pipes should have the following attributes:

- Equal distribution of the part flows

- Equal flow loss coefficient for both lines

- Good velocity distribution at the outlet sections

- No air-pocket formations allowed (this is also valid for the upstream and downstream piping)

- Internal reinforcement, a crescent shaped rib

- Entirely welded construction

Standard design for the horizontal approach is made according to Figure 1(a_1, b_1) and, for the vertical approach, according to Figure 1(a_2, b_2). In countries with large temperature differences from day to night, special designs are needed to compensate the change of pipe length.
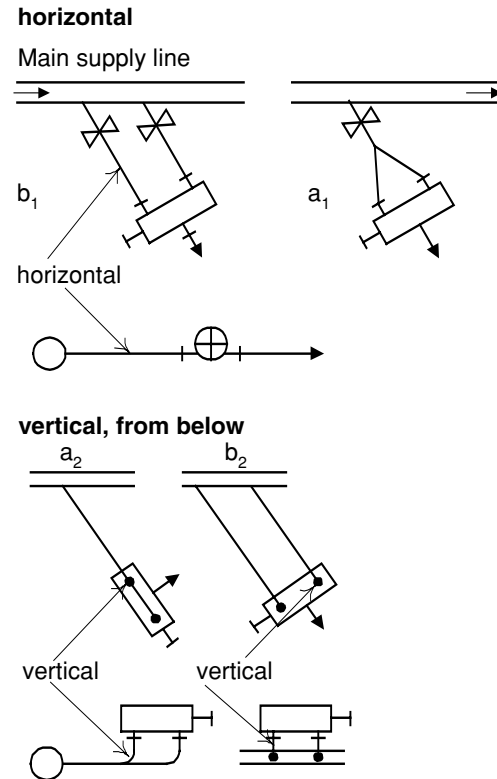

Figure 1. Horizontal Supply Line and Vertical Supply Line from Below.

SELECTION OF WATER TRANSPORT PUMPS

General Criteria

Figure 2 shows the most frequently used impeller arrangements, and Figures 3, 4, and 5 show different pump cross sections for water transport applications. The main feature of pumps handling large size is the horizontal split flange arrangement for easy inspection and simple maintenance. For a special cavern installation, the vertical arrangement of pumps is usually preferred. (This paper does not cover the vertical arrangements.)

Normally the booster pump is designed with a double-flow single-stage impeller in a horizontal or vertical position. The nozzles are positioned below the split flange and are inline. For good suction behavior of the impeller, the flow distribution and the flow acceleration toward the suction nozzles has to be optimized.

The main pump is designed with a double-flow single-stage impeller in a horizontal arrangement, with a multistage double-flow arrangement, or in a single-flow two-stage back-to-back arrangement. The first stage is a specially designed suction impeller, which allows operation of the pump at the best suction conditions and with lowest material losses due to cavitation.

Pump Cross Section

The 1d pump (Figure 3), is a single-stage, horizontally split, double volute, and double suction pump. It is designed for simple maintenance access. After the nuts retaining the casing split flange are removed, the upper half of the casing can be lifted off. The rotor consisting of impeller, shaft, and shaft seals can be checked and lifted out without disturbing the piping connections or pump alignment. The pump can be driven from both sides.

A 2d or 3d pump is shown in Figure 4. This design is selected for higher head applications. A single suction impeller is installed on each side of the double suction impeller. The flow is guided in the diffusers from the first stage to the final stage. The two suction branches are integral to the lower half of the pump casing and are
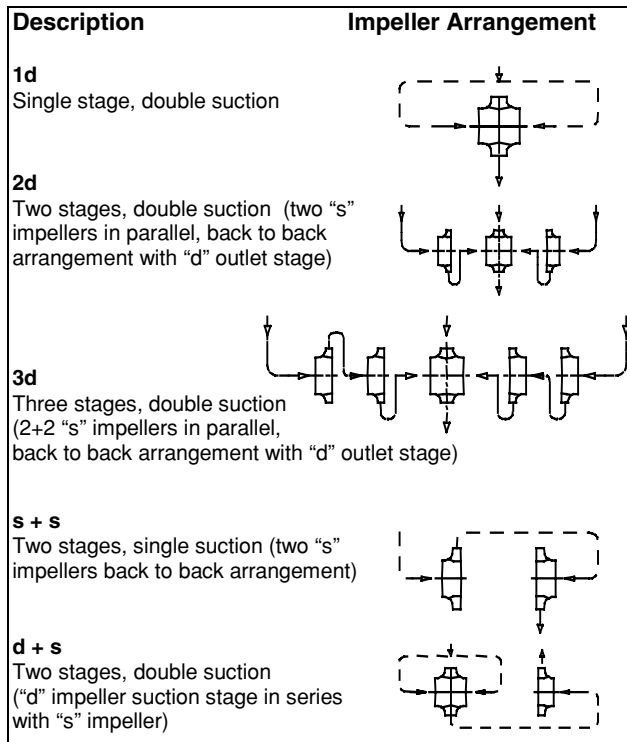
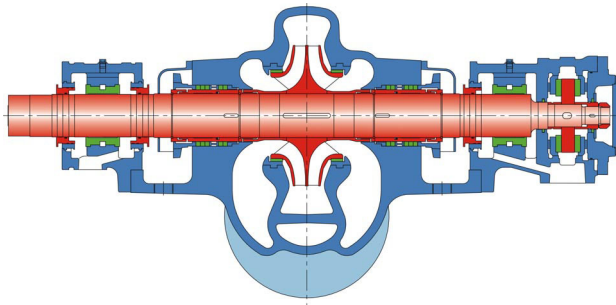| Description	Impeller Arrangement
1d Single stage, double suction	
2d Two stages, double suction (two "s" impellers in parallel, back to back arrangement with "d" outlet stage)	
3d Three stages, double suction (2+2 "s" impellers in parallel, back to back arrangement with "d" outlet stage)	
s + s Two stages, single suction (two "s" impellers back to back arrangement)	
d + s Two stages, double suction ("d" impeller suction stage in series with "s" impeller)	

Figure 2. Most Frequent Impeller Arrangements.

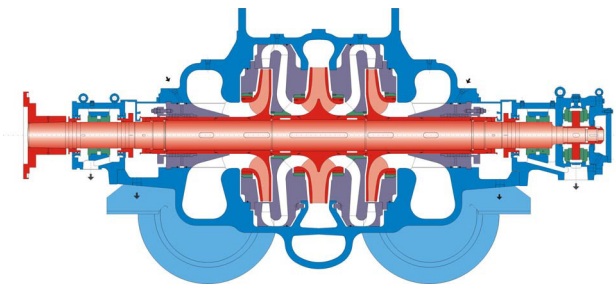*Figure 3. 1d Cross-Section of Pump.*

Figure 4. 2d or 3d Cross-Section of Pump.

connected by means of a breeches pipe. The suction branches can be arranged either horizontally (inline with the discharge) or from below, which results in minimum space requirements, because the suction and discharge piping can be arranged parallel to the pump-motor axis.

The s + s pump, shown in Figure 5, is a single-flow two-stage pump with suction and subsequent impeller in back-to-back arrangement for smaller flow and higher heads. This pump design has an inline nozzle arrangement below the horizontal main split flange. The axial thrust is partly equalized by the back-to-back
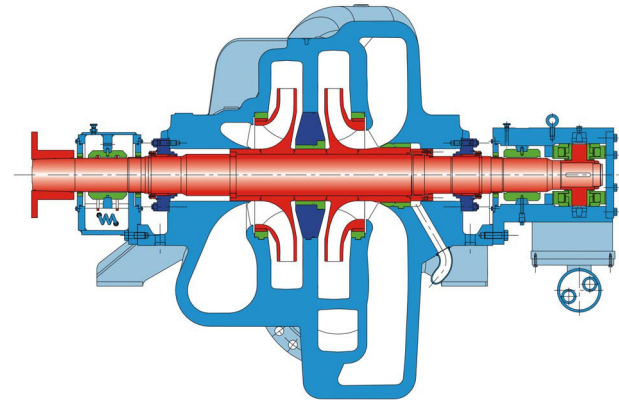

Figure 5. S + S Cross Section of Pump.

arrangement of impellers. The mechanical seal pressure can be equalized with the stage throttle bushing, and sizing of the two bushings (center and throttle) can control the residual axial thrust.

The overall performance range provides for flowrates up to 20,000 m³/h (88,057 gpm) and delivery head up to 680 m (2231 ft). The pumps are designed for the system pressure, with maximum hydrostatic test pressure of 150 bar (2175 psig) or smaller. The maximum allowable fluid temperature range for the water is 4°C to 50°C (39°F to 122°F).

Water Analysis

Where clean water is indicated by the client's specification, the speed limitation is only dictated by the net positive suction head (NPSH) available. The pump material to be employed is cast steel (with surface protection), with the impeller usually made from chromium steel.

If there are solids suspended or dissolved in the water, the speed of the pump has to be limited, as the weight loss due to abrasion will increase with the third power of the relative flow velocity. Consequently, a pump with a lower velocity will always be favored.

The materials are the same as for clean water, but parts subjected to high velocities such as the labyrinths will need to be hard coated to avoid galling and wear between stationary and rotating seal elements. The material selection has to be done to cover different operating conditions (standstill and normal operation condition).

Suction Condition

The available NPSH for water transport pumps depends on the pump and minimum reservoir levels. When lowering the pump foundation is required to achieve better suction conditions, this is associated with high civil works cost. In those cases it is proposed to install booster pumps, which are flooded at the minimum reservoir levels. For those low suction conditions, a booster pump in combination with the main pump is the appropriate selection. The available NPSH limits the speed of the pump. The lower the speed, the larger will be the impeller diameter and, consequently, a larger, heavier and more costly pump. To reduce the overall investment cost, it may pay to use either a low speed booster coping with the low NPSH feeding into a faster running main pump, or use a slow running main pump with huge dimensions, resulting in a heavy pump. For a final proposal it is worthwhile to compare the two different arrangements to obtain reliable suction conditions with low energy consumption.

In the project phase, often the available NPSH value can be proposed by the pump manufacturer to ensure trouble-free operation.

The following general criteria are used. Available NPSH greater than NPSH at 3 percent head drop is required to ensure operation without measurable impairment of head throughout the entire range of operation—in particular at maximum flow. A safety factor has to be used on top of NPSH at 3 percent head drop values in

order to meet measuring tolerances and possible variation of the available NPSH. Noise and vibration caused by cavitation must be limited to an acceptable level, which depends on pump type, service, and client's specifications.

Cavitation erosion must be limited in a way to achieve the guaranteed impeller life or—in the absence of formal guarantees—to meet customers' expectations. A risk of erosion is usually present only in cold-water services at high suction eye tip speeds. But pumps with a nonuniform approach flow or torch-like vortices initiating in the intake, pumping degassed cold-water are at risk of erosion even with moderate suction eye tip speed.

Correct pump selection needs experience, which cannot be laid down with a few rules. The same is true for the definition of the available NPSH required to ensure trouble-free operation. There are many approaches possible. For example, the manufacturer may use bubble tests or acoustic cavitation noise measurements to estimate the material losses due to cavitation in combination with the selected impeller material.

For the majority of applications, the available NPSH for continuous operation must exceed that at 3 percent head drop by an adequate safety margin in order to avoid a loss of performance, noise, and vibrations, or even cavitation erosion. The required safety margin:

● Increases with rising peripheral speed at the impeller inlet

● Is reduced if cavitation-resistant material is used

● Rises with increasingly corrosive media

● Depends on the operating conditions of the pump and the type and temperature of the medium being pumped

Figure 6 shows the approximate values for selecting plant NPSH as a function of NPSH at 3 percent head drop. Depending on the application, deviation can be made from this curve, when slightly higher margins are employed.
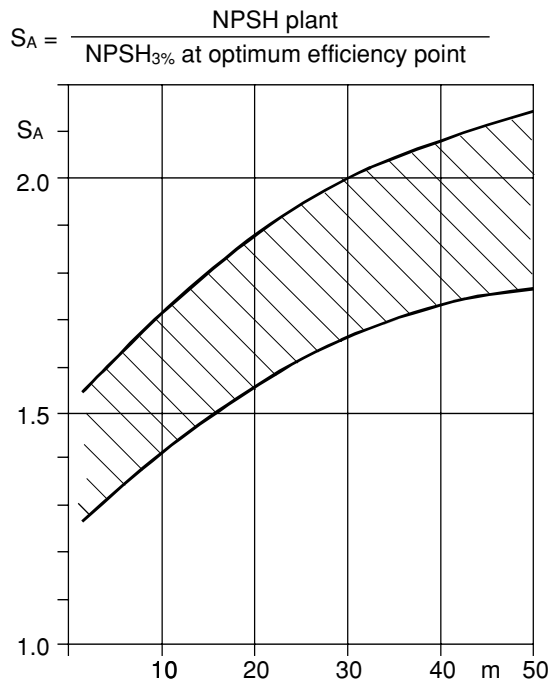
$$S_A = \frac{\text{NPSH plant}}{\text{NPSH}_{3\%} \text{ at optimum efficiency point}}$$

Figure 6. Safety Margin in Relation to NPSH—3 Percent Head Drop.

Pump Efficiency

The choice of pump type has become of particular significance due to the sharp rises of energy costs and the power penalty thereby induced.

The specific speed and, hence, the efficiency reached can be modified using pumps with more stages of splitting the flow, which improves the efficiency. The flow layout, in addition, has a decisive influence on the efficiency of the pumping set. The maximum possible efficiencies for various designs are given in Figure 7. The following pump types are to be considered best for drinking water supply systems:

● Single or multistage, double-flow pumps of designs 1d, 2d, 3d

● Two-stage, single-flow pumps with back-to-back impeller arrangement (s + s)

● The s + s arrangement can also be carried out with a double-flow first impeller, by accepting that a complicated casting technique is thus made unavoidable (d + s).

Relative efficiency at best point η/η₀₀
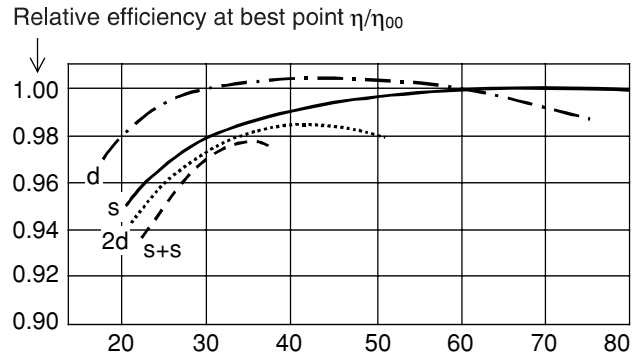
Nq/side = $\frac{n\sqrt{Q}}{H^{3/4}}$ where Q (m³/s), H (m), n (1/min)

η/η₀₀ = for η = 90% (at nq = 70)

Drawn for H_stage = 100 m, impeller diameter 0.5 m

s	single-flow + single-stage + single volute
s+s	single-flow + multi-stage back-to-back arrangement
d	double flow + single-stage + single volute
2d	double flow + two-stage

Figure 7. Hydraulic Efficiency as Function of Specific Speed and Pump Design.

When evaluating the efficiency, the losses brought about by an increase in the wear ring clearance, which is a function of wear, have to be taken into account. Where the efficiency is of greater importance, preference can be given to a two-stage, double-flow pump, rather than to a single-stage, double-flow type, in spite of the former's higher purchase price.

Where the 2d or 3d design is not necessary in order to meet the NPSH requirements, the s + s configuration is to be preferred for the following reasons. The specific speed of rotation is some 40 percent higher. This results in a higher efficiency, in spite of the internal losses due to leakage between the stages.

The distance between the bearings of the machine is shorter and thus gives better conditions as regards rotordynamics, i.e., critical speed. Additional "bearing rigidity" of the clearance between the stages provides further damping. A check of the clearances is possible, when the rotor is in place.

It should, however, be noted that the s + s design requires a significantly higher development effort, i.e., costs, than the 2d or 3d as regards casing design, hydraulic layout of the diffusers, axial pressure equalization, mechanical design, and radial thrust.

A final decision regarding the type of pump to be chosen can be made by the pump manufacturer and the plant design engineer after careful consideration of the advantages and disadvantages previously mentioned, taking into account also the fluid pressure at

the impeller inlet, the efficiency, the space occupied by the pump, and the civil engineering measures to be taken.

Evaluation of Pump Efficiency

When designing water transport pumps, particular attention has to be paid to achieving low operating costs, as the operating costs predominate over the purchase price. When calculating the economic performance, it is necessary to take the efficiency of the pumps into account.

The energy penalty for a large pump is calculated below considering the additional consumption of energy brought about by a lower efficiency.

A comparison of single and multistage, double-flow water transport pumps with the same delivery head and approximately nq = 30 (specific speed) is used as the basis. The efficiency rises with increasing impeller size. This is brought about by the fact that greater pumping capacity is achieved by increased impeller diameter, which in turn requires a greater flow cross section and reduces the effect of the boundary layer. Further it was assumed that the average operating time per year of the pumps amounted to 4000 hours. The standby pump, which is not normally operating, was included in the calculation. Assuming, as an example, a 0.5 percent drop in efficiency, when 89.5 percent instead of 90 percent is reached at 9500 kW rating, the pump power consumption is 90/89.5 × 9000 = 9553 kW (i.e., the power loss, which is subject to a penalty, amounts to 53 kW). In this case the price of 1 kWh is assumed to be $0.10. Additional energy costs are (G) = increased power consumption × operating hours per years × operating life × price per kWh. Where a 0.5 percent efficiency drop occurs, the higher operating costs have to be paid during the whole operating life of the pump (n). These costs have to be covered by a suitable financial provision at the time of commissioning. Assuming the rate of interest to be 10 percent per year (interest multiplication factor R = 1.1), a capital provision (C) has to be made at the time of pump commissioning in order to pay for the efficiency loss of the pump over time.

$$C = \frac{G}{n \bullet R^n} \bullet \frac{R^n - 1}{R - 1} \qquad (1)$$

$$C = \frac{53 \bullet 4000 \bullet 20 \bullet 0.1}{20 \bullet 1.1^{20}} \bullet \frac{1.1^{20} - 1}{1.1 - 1} = \$180,487 \qquad (2)$$

Consequently the penalty is approximately $3400/kW for the above example.

The pumping station designer can influence to some extent the type of pump required by distributing the total delivery flow over the appropriate number of pumps. The highest possible efficiency is to be obtained by suitable choice of the number of pumps and of the speed of rotation. There is an illustration of the relative peak efficiency for several pump types in Figure 7.

Polishing the impeller flow channels has an influence on the efficiency of water transport pumps. The costs of improving the efficiency this way are low in comparison with those of the complete plant.

An example of the approximate cost distribution for a water transport system in the Middle East is given here. It has to be noted that the specific energy requirement in kWh/m^3 is 10 times the value that applies for a normal European installation, since the pipeline lengths are much greater.

- Costs of pipeline: 37 percent
- Costs of pumping station: 20 percent
- Operating costs of pump for 20 years: 42 percent
- Costs of pumps and baseplates: 1 percent

## PUMP MECHANICAL CONSIDERATIONS

Axial Thrust

In the case of the 1d, 2d, or 3d arrangement, the static axial thrust is almost equalized since the impellers are symmetrically arranged. The dynamic part of the axial thrust occurring at part load is small and can be taken up by a double-acting, axial thrust bearing. A big advantage of this design is the fact that the axial thrust scarcely changes even when the clearances are worn, due to the symmetrical arrangement of the impellers.

The axial thrust conditions of the s + s design are, as expected, completely different. Due to the arrangement of the impellers, the pressure conditions behind the individual stages are not the same. In addition, leakage flows occurring in various directions within the side spaces at the rear of the impellers cause a further change of the pressure distribution.

A typical example for both machine types can be seen in Figure 8. It is to be expected that an increase of clearance in an s + s machine brings a further change of axial thrust, due to reasons already explained. Reliable values can be derived only by measurements on model machines so that the magnitude of the axial thrust bearing, which is larger than that of the 2d or 3d design, complies with the requirements applying for all operating conditions.
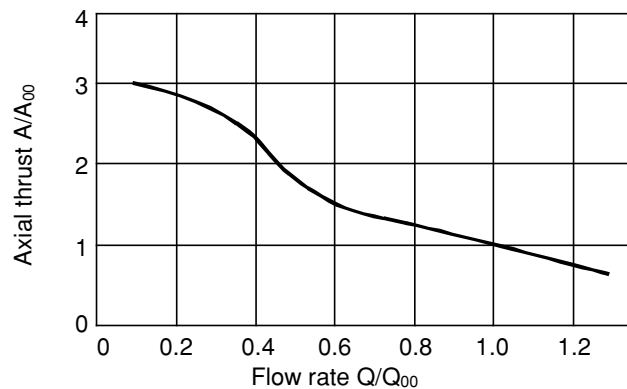
a Dimensionless axial thrusts as a function of the flow rate for back-to-back s+s arrangement
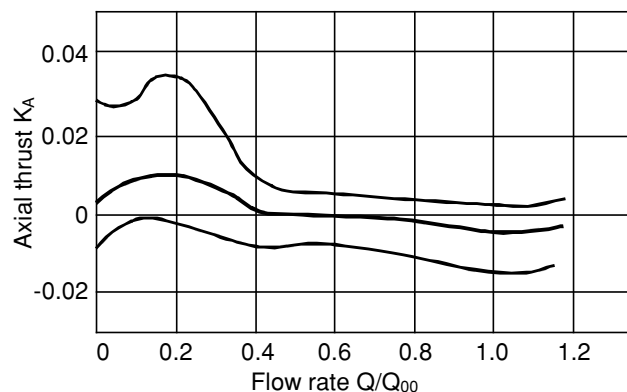
b Dimensionless axial thrusts as a function of the flow rate for two stage, double-flow 2d arrangement

Figure 8. Axial Thrust Measurements on Horizontally Split, Multistage Pumps of Various Design.

Bearings and Lubrication

Forced feed or self-lubricated hydrodynamic bearings are used. They consist of a journal bearing unit on the drive-end and a combined thrust/journal bearing on the nondrive-end. The thrust

bearing absorbs the residual thrust of the hydraulically balanced rotor. The bearing housings are horizontally split for ease of assembly and maintenance. Antifriction bearings are available for smaller pumps, i.e., booster pumps.

The lubrication of the sleeve bearings requires a forced feed lube oil system. For large pumps, a mechanically driven main lube oil pump is flexibly coupled to the main shaft. The electrical driven startup lube pumps are installed into the lubricating oil reservoir. Attention has to be paid in cases where the pump may run in reverse due to incorrect operation.

Self-contained equalizing bearing units can be used up to a limiting shaft size. Within these limits they can function effectively. System lubrication is autonomous, provided by means of a viscosity pump (thrust collar) driven by shaft rotation. This pump consists of the combination of the built-in mechanical oil circulator and the rotating collar, which draws oil up for the sump as soon as the shaft begins to rotate. As long as the shaft rotates, pumping action is achieved. No priming is necessary since the pump inlets are always submerged. Pumping action is bidirectional and automatically adjusts with shaft rotation. The pressure and flow generated by this pump force the oil out through passages in the housing to lubricate both thrust bearings and the internal journal bearing. The pressure in the thrust cavity then drives the hot oil through the oil cooler and than back into the oil sump. Sufficient flow and pressure are developed by this pump to send oil to a separate journal bearing if required. The system eliminates the need for an electric motor/pump, emergency pump, accumulator, run-down tank, or other special arrangements for oil lubrication. No additional motor control system or any electric power is required to circulate the oil. And since no separate lube system is required, the cost savings are significant. Attention should be given to ensure that the oil system could operate when the shaft rotates at the minimum continuous operating speed. Before starting up the pump, the journal bearing should be lubricated once manually. When connecting different bearings, the supply line should be adjusted so that the required oil flow is established and the drains are connected to the main oil sump.

Rotordynamics

Increasing machine costs force the manufacturers to continuously increase the power concentrations in their pumps. This increase in power concentration leads to vibration induced problems with pumps. Today, there is the need to calculate the critical speed and, much more importantly, the limit of stability.

Figure 9 shows why API 610, for example, prohibits operation within certain limits of a critical speed. With little damping, the vibration amplitudes are much too large. The second curve in the diagram indicates qualitatively the experience with pumps. Rotor damping is more than large enough to prevent any resonant amplification, and, accordingly, critical speeds can barely be detected at the machine.

This is presented in detail with the Campbell diagram shown in Figure 10, where the Eigen-frequencies are plotted versus machine speed. As the pump rotor has continuous mass distribution, in contrast to a single mass system, there are theoretically an infinite number of Eigen-frequencies.

Today, several modern rotordynamic calculation tools are available. Rotors are modeled with state-of-the-art finite element computer codes specially designed for rotordynamic calculations. Oil lubricated journal bearings are introduced as a spring-damper boundary element, using measured coefficients from the manufacturer. The influence of annular seals at the impeller inlet, between single entry stages and at interstage balance sleeve, is computed based on the theory of Childs (1982). Impeller-casing interaction is evaluated using inhouse measurements and, together with the annular seals, is introduced as an additional spring-damper-mass boundary element.
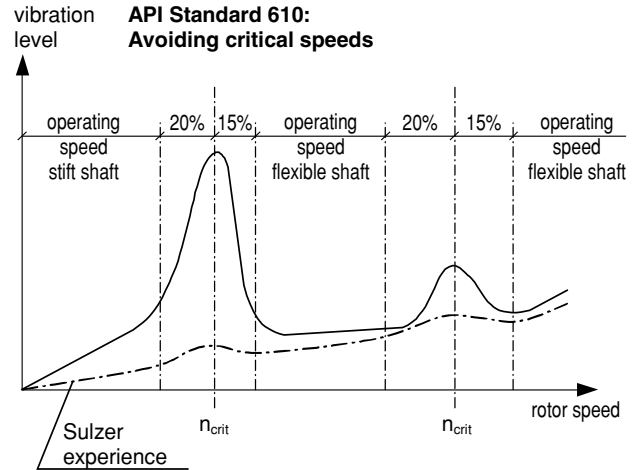

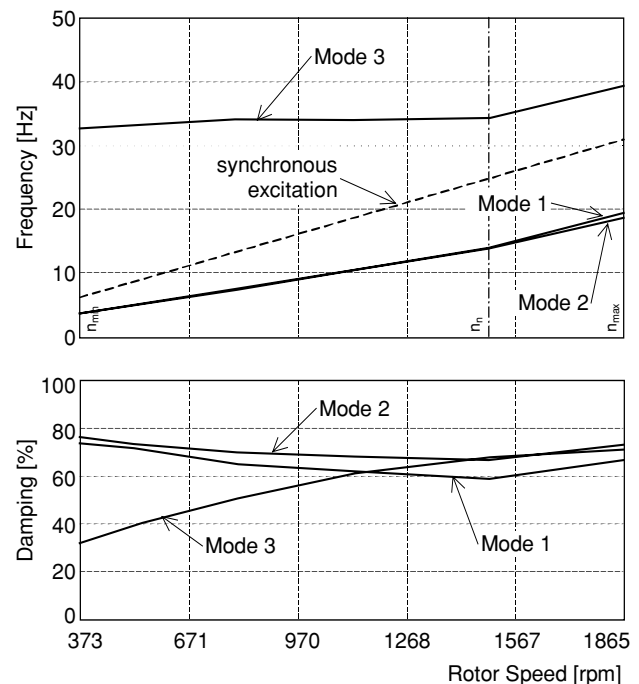Figure 9. *Separation from Critical Speed in Accordance with API 610.*

Figure 10. *Campbell Diagram—Eigen-Frequency Versus Rotor Speed.*

The following statements can be made for the different pump types (Figure 11):

● *Single-stage, double entry machine*—As the bearing span is very short, the total stiffness of the rotor is defined mainly by the shaft diameter and the journal bearings. The stiffness of the only two short annular seals, which are relatively small compared to the stiffness of the shaft, have very little influence on the Eigen-frequency. An increase in the seal clearance, hence, cannot change Eigen-frequency significantly. Furthermore, antidamping forces are not to be expected, since the layout with double volutes has relatively large separation between the tongue and impeller external diameter. When designing a horizontal split machine, having mechanical sealing and suction side configuration capable of accepting the full backpressure, great care has to be taken regarding rotordynamics. This is particularly applicable when a relatively thin shaft is provided in the interest of high suction capacity.
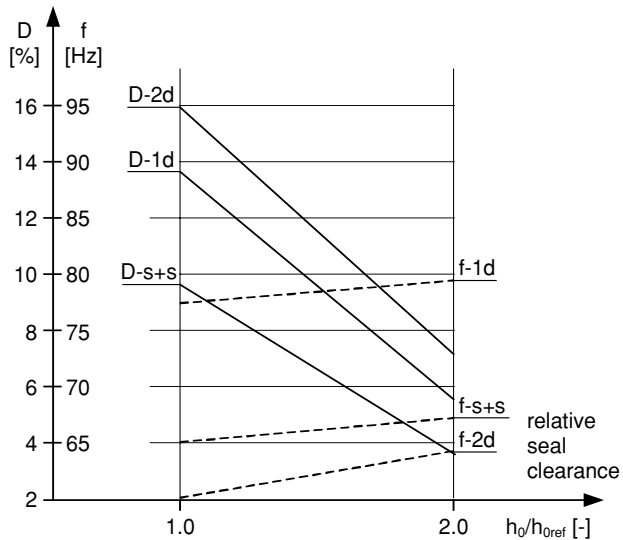
Figure 11. Damping D and Eigen-Frequency F as Function of Annular Seal Wear—Comparison of Different Pump Types.

● *Multistage, double entry machine*—In this case, the bearing span is larger than that for the single-stage pump and, with the same shaft diameter, the slenderness-ratio of the rotor is smaller and the Eigen-frequency is lowered. The impact of the seal clearances on the total stiffness of the rotor becomes greater because of the at least three times larger number of annular seals, but they still do not govern the rotor behavior. Reduction of the seal stiffness by increasing the clearance does not lower the Eigen-frequency significantly. This trend is exhibited by the two cases with one or two stages with relatively few annular seals and no balance sleeve. In the presence of relatively large separation between impeller and diffuser of the first stage, and with a double volute in the second stage, the resulting radial thrust is relatively small.

● *Two-stage, back-to-back machine*—Because of the shorter bearing span and the additional long and stiffening annular seal between the two stages, the Eigen-frequency increases by 30 to 40 percent over that of the 2d machine. However, the damping is evidently lower than in the 2d or even the 1d design. This is the effect of the strongly destabilizing cross-coupled stiffness of a long annular seal subject to water with a large inlet swirl. There is the possibility of reducing the swirling of the water entering the interstage seal. This action can raise the rotor damping of the s + s machine much further than is needed for good rotordynamic performance, and will do so without an efficiency loss. Even with worn seals, the pump will operate stably. The shaft diameter can be reduced with the application of such a swirl brake, resulting in better suction behavior and efficiency.

The shape of the diffuser, i.e., volutes, between both stages has to be carefully considered so that the resulting radial thrust remains small, particularly at part load, and no flexing of the shaft resulting in contact within the clearance occurs.

Maintenance

Water transport pumping stations are often located in countries where well-trained service personnel are scarce. Consequently, ease of pump maintenance has to be assured. Water transport pumps should feature horizontally split casings. These can be easily taken apart without disturbing the suction and discharge flange connections. On account of rising energy costs, the planning engineer will, in the future, have to select those pump types, which entail a minimum of maintenance work. An indication that maintenance is necessary on a given pump is provided by:

● A falloff in efficiency

● A rise in vibration due to wear, e.g., on the annular seals

In view of the demand for minimum maintenance, it is therefore correct to select pumps with high specific speed. As may be seen in Figure 12, the relative leakage loss drops steeply with increasing specific speed. In addition to the selection of the appropriate pump type and size, there is also a need to select the suitable resistant material or coatings, which can guarantee an extended lifetime. For example, surface treatment on annular seals with hard layers is common. There can be no doubt that in the future the decision regarding the choice of pump type to be installed will be influenced to a major degree not only by the purchase price but also by the predicted life of the pump components based on statistical data from the similar pumps gathered from the experience in the field. In addition, the following operating measurements can indicate a problem:

● Vibration levels on the shaft

● Temperature rise in the bearings (indication of a change of the axial thrust because of the wear in the annular seals)

● Temperature rise through the pump (a general indication of efficiency drop because of wear)

They give clear indications of the internal condition of the pump parts. These indications can enable the operator to decide whether a repair of the internal pump parts must take place in order to prevent a breakdown.
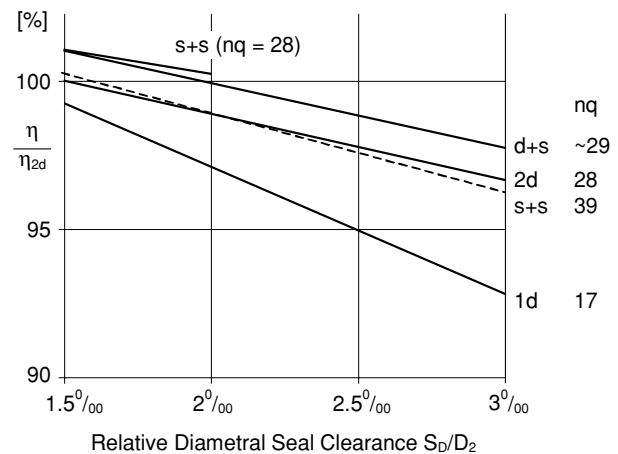

Figure 12. Relative Leakage Losses of Annular Seals at Single Entry and Double Entry Pumps.

PUMP DRIVER SELECTION

The drivers for large water transport pumps are usually three-phase electric motors. A few guidelines are given below to assist in selecting the correct motor for a particular drive. The information concerning the principal applications of asynchronous motors, i.e., squirrel-cage and slip-ring induction motors, and of synchronous motors can only be of a general nature, since in many cases a careful investigation will be unavoidable to meet the supplementary requirements specified. These additional requirements are dependent on the following influencing factors:

● The working machine as the pump

● The ambient conditions

● The electricity supply grid

● The regulations in force

Frequently, these requirements involve oversizing or a special design instead of standardized motors.

Applications for Three-Phase
Asynchronous Squirrel-Cage Motors

Due to its robust construction, the squirrel-cage motor can be regarded as a general-purpose type. It is suitable for a large number of working machines and is utilized in the water transport areas.

In general, a high-speed motor is cheaper for a given torque value. When a gear train is used, the cost of motor plus gear train must be taken into account when determining the appropriate motor speed. A wide range of special versions facilitates the matching of motors to their applications still further. They include:

● Stepped speed designs (pole change motor)

● Specially matched torque characteristics

● Special types of construction and protection

● Infinitely variable speed facility obtained by electronic control via converter-controlled voltage and frequency regulation

When directly connected at full grid voltage, three-phase squirrel-cage motors take up a transient current that can amount to between four and six times rated current, depending on power and the number of poles. During runup the current decreases to a value corresponding to the load, e.g., to rated current when rated load is applied. The starting current can be reduced by special startup procedures.

These motors are designed for direct engaging. Current and torque are valid for startup at nominal voltage. When the counter-torque so permits, a motor suitable for the impedance of part voltage starting or for starting via a block transformer can also be supplied. When starting at reduced voltage torques, decrease approximately in accordance with the square of the voltage and the current roughly linear to it.

The speed of these motors can be changed in steps by pole switching or continuously by frequency changing. Frequency change by means of converter or transformer usually has a control range of 1:5 and above.

Applications for Three-Phase
Asynchronous Motors with Slip-Ring Rotors

The slip-ring rotor type requires greater expenditure both for purchase and in operation (e.g., servicing of slip-rings and brushes). It is therefore used for those applications where the simple squirrel-cage motor is inadequate. Its advantages are to be found in the following properties:

● Low starting current

● High starting torque

● Easy speed changing (loss-free or dissipative)

● Permits a high switching frequency

In the case of slip-ring induction motors, additional resistance inserted into the rotor current circuit serves to raise the torque during startup and also to increase breakdown slip. In general, the resistances are so chosen that it is possible to startup from standstill with rated torque (which corresponds approximately to rated current). At the time, the additional resistances limit the starting current of the slip-ring induction motor by comparison with that of the squirrel-cage motor.

Starting takes place via starting resistances inserted in the rotor current circuit. This increase of the resistance in the rotor current circuit causes a raised starting torque that is approximately equal to breakdown torque. The starting heat is dissipated essentially via the starting resistance so that motor stress during acceleration is slight.

The following methods are used for speed variation of these motors:

● Control by means of resistance in the rotor circuit. The control causes losses (slip losses) and is therefore usable to a limited extent only (poor efficiency)

● Control by means of subsynchronous converter cascade (or static slip energy recovery system). This is particularly economical when the speed adjustment range is limited. The slip output is fed back into the three-phase grid. Hence, the control is low-loss, with good efficiency, and is applicable for outputs of up to and more than 10 MW. The economic speed adjustment range is limited to about 50 percent of the nominal speed.

Applications for Three-Phase Synchronous Motors

Unlike the asynchronous motor, the synchronous type requires no reactive power for its excitation. Depending on the field current value, the synchronous motor can even deliver reactive power to the electricity supply grid. The synchronous motor with power factor 1.0 will achieve approximately 0.7 percent higher efficiency than the squirrel-cage motor.

Startup of synchronous motors usually takes place asynchronously. The shape of the speed/torque characteristic is dependent on the design of the pole shoes and the damping winding. The design referred to above produces an almost constant torque over the whole speed range. In the case of direct engaging, a transient current occurs as with asynchronous motors. This can be decreased by suitable startup procedures. Connecting the direct current excitation brings about change over to synchronous speed.

The speed control of three-phase synchronous motors is designed to operate at fixed speed, with constant voltage and frequency. In special cases, speed control is achieved by feeding the motor with variable frequency and voltage by means of static frequency converters. This circuitry is expensive, especially in the higher MW range.

EXAMPLE OF SIMPLIFYING A PUMP HYDRAULIC LAYOUT

In a pipeline system, the interaction of the different pumping stations has to be considered when selecting the optimal hydraulic layout.

Figure 13 shows a simple system, which has two operating phases (initial and final [Table 1]) with pronounced geodetic head (high points) in the first part of the pipeline. In this case, the system planner and the pump manufacturer came to the conclusion of proposing the following pump combination.
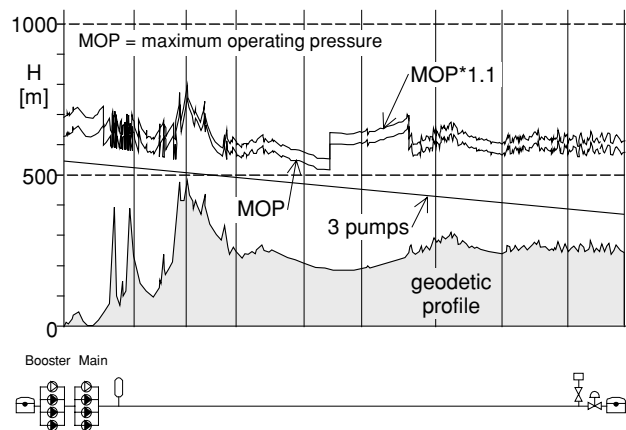

Figure 13. Pipeline Profile with Stationary Operating Condition Case 1 (Booster, Main Pumps).

A booster pump is used at an appropriate speed to fulfill the suction condition of the plant and to provide sufficient suction head for the main pump. With a two-stage main pump, an additional

Table 1. Operating Conditions of Case 1.

Operating condition	Initial condition	Final condition
Total Flow m³/h (USGPM)	18'942 (83'400)	34'098 (150'132)
Unit split	3 (+1 standby)	6 (+2 standby)
Quantity per unit m³/h (USGPM)	6'314 (27'800)	5'683 (25'022)
Booster pump	Single stage	Single stage
Speed (RPM)	742	742
Specific speed nq (ns)	32.5 (1'680)	32.5 (1'680)
Head per stage m (ft)	57.5 (188.6)	61.2 (200.8)
Power kW (BHP)	1'096 (1'470)	1'049 (1'407)
NPSHav / NPSHreq m (ft)	7.8 (25.6) / 5.3 (17.4)	7.8 (25.6) / 4.6 (15.1)
Impeller diameter mm (in)	905 (35.63)	905 (35.63)
Impeller arrangement	Double flow	Double flow
Main Pump	Two stage	Two stage
Speed (RPM)	1'490	1'490
Specific Speed nq (ns)	35 (1'550)	30 (1'550)
Head per stage m (ft)	243.25 (798)	272.4 (894)
Power kW (BHP)	9'353 (12'542)	9460 (12'686)
NPSHav/ NPSHreq. m (ft)	65.3 (214) / 25.1 (82.3)	69 (226) / 23.4 (76.8)
Impeller diameter mm (in)	928 (36.5)	950 (37.4)
Impeller arrangement	Back-to-back, single flow	Back-to-back, single flow

(…) Values in bracket belonging to the US-units

pumping station half way to the final high point of the pipe-line could be avoided, saving the investment cost of supplying electric power and other services to this intermediate station. The negative aspect was the higher pipe pressure up to the position where the second pumping station would have been built. The proposed concept has reduced the investment cost, and a good combination of booster and main pump has optimized the operating cost.

Figure 14 shows a more complicated closed system, which has to feed variable flows over large distances with a slowly rising geodetic head with typically no high point. The system planner proposed four pumping stations with parallel operating pumps in the first pump station (booster and main) and series operating main pumps in the second and fourth stations, which can be bypassed during low flow periods. The third pump station is equipped with parallel operating pumps of the same size as in the first station. With this combination, the number of different pump types has been reduced to three pump types: booster (1d, double-flow pump), first and third main pump (s + s, two-stage back-to-back pump), and second and fourth main pump (1d, single-stage double-flow impeller that has the capacity of the full station).

It is important to use the booster pump at the appropriate speed to fulfill the available suction conditions of the plant and to provide sufficient suction head for the main pump. Further, the interme-diate second and fourth pumping stations can be bypassed at low flowrates. The proposed concept has given the system maximum flexibility with optimized operating cost.

Figure 15 shows an open system, which is operating with fixed speed motors and a reservoir half way along the pipeline, followed by the highest geodetic height of the pipeline profile. The system planner proposed three pumping stations. The first with a parallel operating booster pump (vertical axis) and the remaining two stations with parallel operating horizontal double-flow three-stage pumps of identical size. With this combination, the number of different pump types has been reduced to a booster pump and main
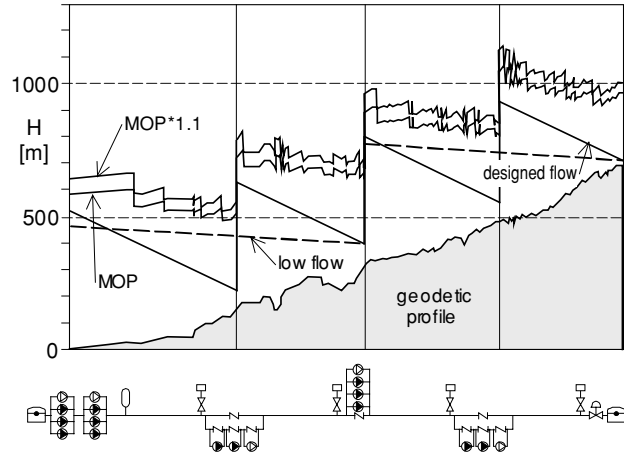

Figure 14. Pipeline Profile with Stationary Operating Condition Case 2 (Booster, Main Pumps Parallel + Series).

pump. It is important to use the booster pump at the appropriate speed to fulfill the available suction condition of the plant and to provide sufficient suction head for the main pump. Further, the second intermediate pumping station has its suction condition determined by the reservoir. The proposed concept takes in account the simplicity of the pump.
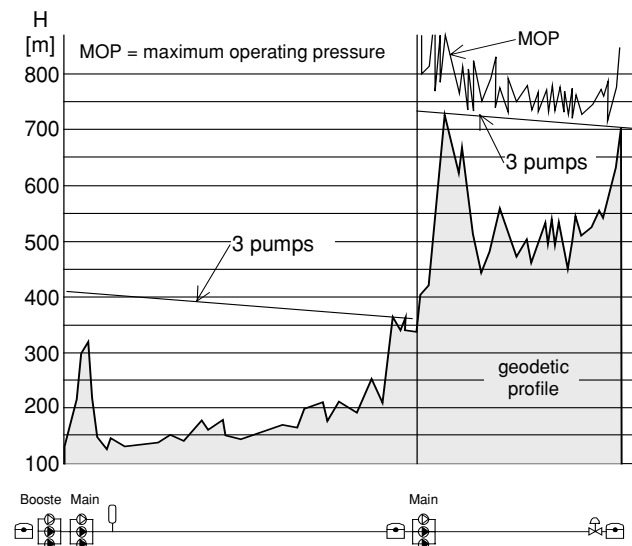

Figure 15. Pipeline Profile with Stationary Operating Condition Case 3 (Open System).

Matching the Water Demand

The required flow and head of drinking water pumps are obtained from the size of the town and the height difference between water source and location of the consumer. The water demand fluctuates seasonally, monthly, daily, and hourly. The hourly water consumption varies during one day by a factor of approximately one to three. Depending on the relation between domestic and industrial water consumption, the present-day specific demand rate varies in civilized areas between 150 and 500 l/day (39.6 and 132 g/day) (liters per head of population and day). The peak demand rate, however, can be much higher. This illustrates that water transport pumps have to be capable of delivering water over a wide operating range. In order to vary the flow in accordance with the demand, this requires a number of units operating in parallel to adjust the flowrate.

The three projects give good examples, which cover optimum selection and realistic number of units to split the power into an acceptable power range. All systems have very low available NPSH-values, so that booster pumps have to be supplied to achieve good suction conditions for the main pumps, which can then operate within an optimum specific speed range of 30 to 36. The power demand is in the range of 8000 to 10,500 kW per machine.

The curves are shown for the following cases:

● In Case 1 (Figure 16), the main pumps for the initial phase are built with reduced impeller diameter and will be changed, in the final phase, to a larger impeller diameter set. The booster pumps will remain the same. The s + s machine is capable of coping with both operating conditions. The high head of approximately 500 m (1640 ft) is split into two stages. The main pump has a suction impeller, which is designed to run for 40,000 hours with optimum suction behavior. The pumps are running with constant speed and, for starting purposes, a station bypass on the booster and main station and can be used for limiting the main flow. As an example, the main pump performance curve is shown in Figure 17 for the initial flow conditions.
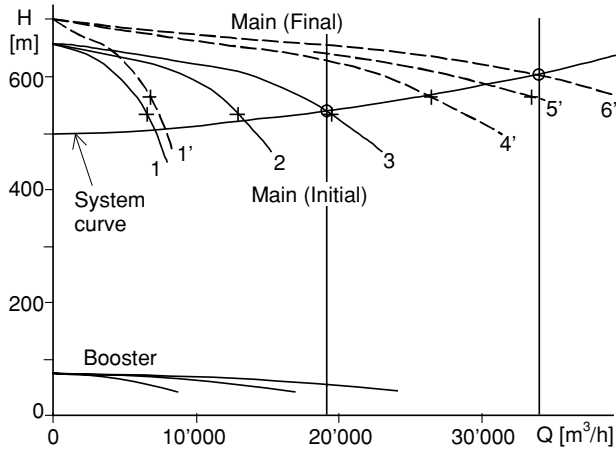

Figure 16. Case 1 Booster and Main Pump for Initial and Final Phase. All Pumps Are Running in Parallel.
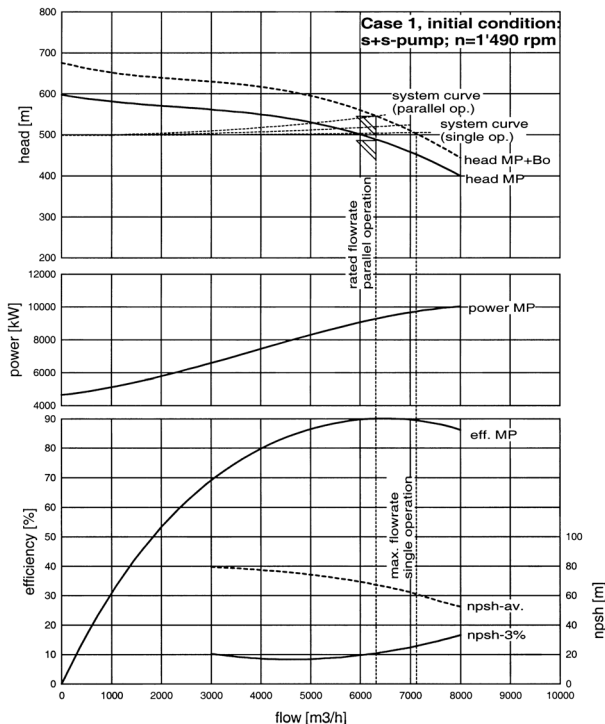

Figure 17. Case 1 Main Pump Performance Curve for Initial Phase.

● In Case 2 (Figure 18, Table 2), the booster and main pumps are operating normally in a parallel combination, but the intermediate pump station is added in series. This system uses variable speed main pumps throughout, whereas the booster pumps are running at constant speed.
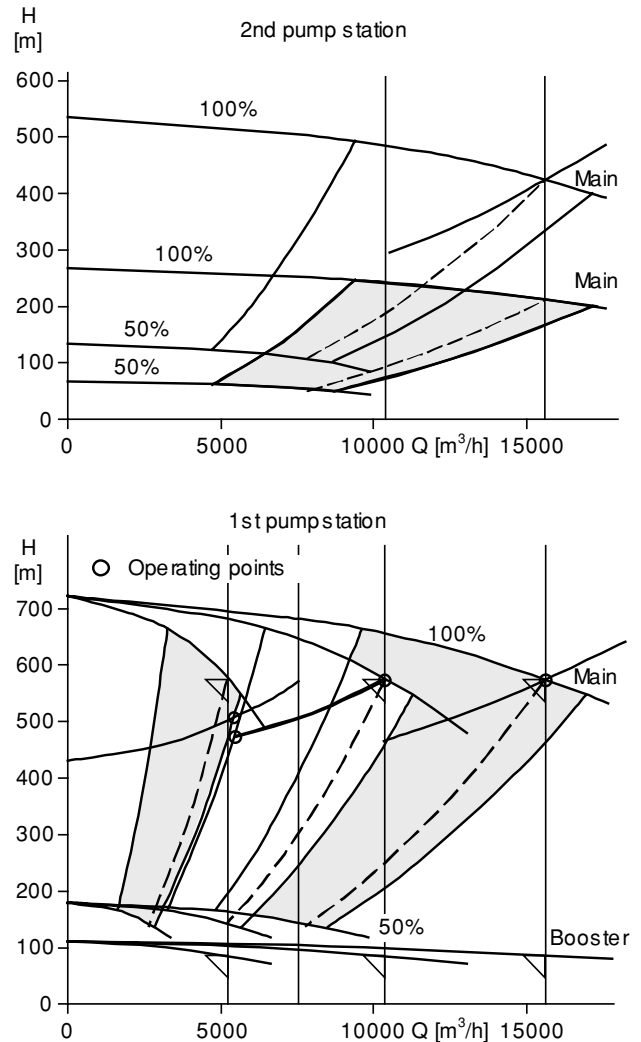

Figure 18. Case 2 Booster and Main Pump in First Station Operating in Parallel and in Intermediate Second Station in Series.

● In Case 3 (Figure 19, Table 3), the booster and main pumps are operating in parallel, whereas the main pump in the second station has a slightly different flow friction head curve.

The hydraulic layout of the main pumps has been mainly based on model pump performance tests.

DRIVERS FOR THE THREE CASES

The specification and supply of the electrical equipment were not in the pump manufacturer's scope of supply. The main issue is the starting of such large machines on a relatively limited electric network. The pump manufacturer has to provide the starting torque curve and the inertias. The electric motor manufacturer and the supplier of the high voltage transformers from the grid have to decide and to evaluate the possible starting procedure for the whole group.

● *Constant speed driver with soft starter*—In Case 1, due to the pipeline profile, it has been agreed to use a soft starter to accelerate the motor in the first few seconds against a closed discharge valve and then open partially the main throttle valve. The soft starter will limit the smooth acceleration of the equipment to the full speed in

Table 2. Operating Conditions of Case 2.

Operating condition	Single unit	Total operating units
Total Flow m³/h (USGPM)		15'840 (69'744)
Unit split	1	3 (+1 standby)
Quantity per unit m³/h (USGPM)	5'280 (23'248)	5'280 (23'248)
Booster pump (parallel) 3 (+1)	Single stage	Single stage
Speed (RPM)	715	715
Quantity per unit m³/h (USGPM)	5'280 (23'248)	5'280 (23'248)
Specific speed nq (ns)	28 (1'446)	28 (1'446)
Head per stage m (ft)	75 (246)	75 (246)
Power kW (BHP)	1'212 (1'625)	1'212 (1'625)
NPSHav/ NPSHreq m (ft)	7.8 (25.6) / 3.5 (11.5)	7.8 (25.6) / 3.5 (11.5)
Impeller diameter mm (in)	995 (39.2)	995 (39.2)
Impeller arrangement	Double flow	Double flow
Main Pump (parallel) 3 (+1)	1st station (Two stage)	3rd station (Two stage)
Speed (RPM)	1'740 ÷ 1'140	1'740 ÷ 1'140
Quantity per unit m³/h (USGPM)	5'280 (23'248)	5'280 (23'248)
Specific Speed nq (ns)	33.4 (1'726)	33.6 (1'736)
Head per stage m (ft)	251 (823)	249 (817)
Power kW (BHP)	8'106 (10'870)	8'042 (10'784)
NPSHav/ NPSHreq. m (ft)	62.0 (203) / 25.5 (83.7)	62.0 (203) / 25.5 (83.7)
Impeller diameter mm (in)	815 (32.1)	806 (31.7)
Impeller arrangement	Back-to-back, single flow	Back-to-back, single flow
Main Pump (series) 2 (+1)	2nd station (single stage)	4th station (single stage)
Speed (RPM)	1'170 ÷ 770	1'170 ÷ 770
Quantity per unit m³/h (USGPM)	15'840 (69'744)	15'840 (69'744)
Specific Speed nq (ns)	32 (1'653)	32 (1'653)
Head per stage m (ft)	212 (696)	212 (696)
Power kW (BHP)	10'061 (13'492)	10'061 (13'492)
NPSHav/ NPSHreq. m (ft)	62.0 (203) / 22.0 (72)	62.0 (203) / 22.0 (72)
Impeller diameter mm (in)	1'080 (42.5)	1'080 (42.5)
Impeller arrangement	Double flow	Double Flow

(…) Values in bracket belonging to the US-units

approximately 30 seconds. This arrangement is possible when the operating range is limited because of small level fluctuations and a flat system curve. The filling up of the system has to be made with a mobile pump. The soft starter nearly doubles the motor costs. There are no efficiency losses during full speed operation.

• *Variable speed frequency controlled motor*—In Case 2, the variation of flow was a large issue, because the system curve was friction dominated. The flow and the suction head downstream of the next pumping station needs to be controlled by the speed to avoid low suction heads. When using a frequency-controlled motor, the optimum power consumption can be reached by running the pump close to the best efficiency point. This electric equipment is very costly.

• *Constant speed synchronous motor*—In Case 3, the synchronous motor has the highest motor efficiency and can adjust also to the power factor in the network. The operation of such a motor requires optimum design of the starting process to avoid any
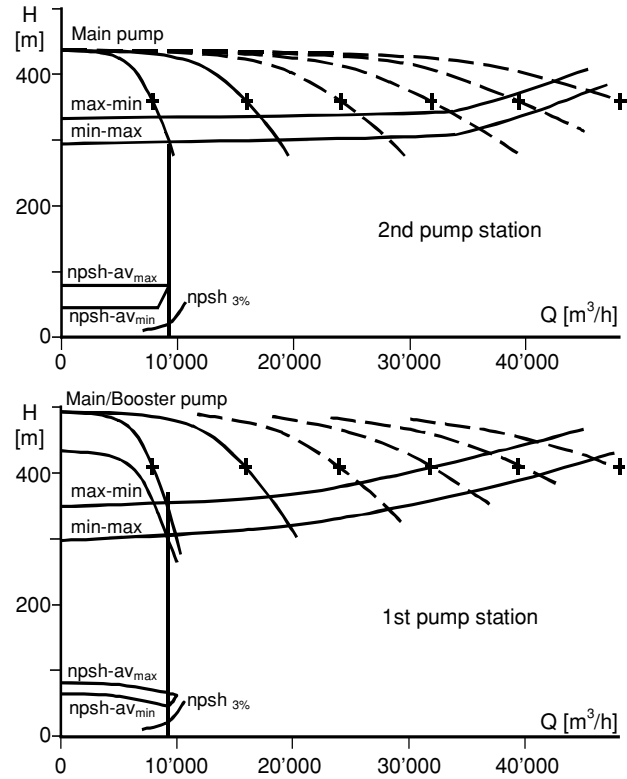

Figure 19. Case 3 Booster and Main Pump in First and Second Station Operating in Parallel.

Table 3. Operating Conditions of Case 3.

Operating condition	Initial condition	Final condition
Total Flow m³/h (USGPM)	17'280 (76'080)	40'680 (179'100)
Unit split	2 (+1 standby)	6 (+2 standby)
Quantity per unit m³/h (USGPM)	8'640 (38'040)	6'780 (29'850)
Booster pump	Single stage	Single stage
Speed (RPM)	596	596
Specific speed nq (ns)	36 (1'860)	36 (1'860)
Head per stage m (ft)	47.5 (1855.8)	53.0 (174.0)
Power kW (BHP)	1'225 (1'643)	1'049 (1'407)
NPSHav / NPSHreq m (ft)	13.0 (42.6) / 5.7 (18.7)	13.0 (42.6) / 5.9 (19.4)
Impeller diameter mm (in)	1'020 (40.2)	1'020 (40.2)
Impeller arrangement	Double flow	Double flow
Unit split	2 (+1 standby)	6 (+2 standby)
Main Pump	Three stage	Three stage
Quantity per unit m³/h (USGPM)	8'100 (35'663)	6'780 (29'850)
Speed (RPM)	1'200	1'200
Specific Speed nq (ns)	36 (1'860)	36 (1'860)
Head per stage m (ft)	117.1 (384.1)	129.3 (424.3)
Power kW (BHP)	8'500 (11'399)	7'875 (10'561)
NPSHav/ NPSHreq. m (ft)	48.0 (158) / 19.5 (64)	46.0 (159) / 12.5 (41)
Impeller diameter mm (in)	810 (31.9)	810 (31.9)
Impeller arrangement	Double flow	Double flow

(…) Values in bracket belonging to the US-units

unacceptable starting overloads. The flow has to be throttled to adjust the power loading during startup of the units.

DESIGN APPROACH FOR ENGINEERED PUMPS

The recent project described in Case 1, which has passed the commissioning phase, shows the approach of optimal design of an engineered pump.

• Based on the model pump and prototype results of the hydraulics used for Case 2, the scaling of the diameter was made, whereas the impeller was scaled and the volutes selected to meet the initial and final flowrates. The suction impeller inlet shape was optimized with a numerical prediction method of cavitation that can determine the bubble distribution from part load to overload flow condition for different suction conditions. With these values, the appropriate material could be selected and the material losses to guarantee the impeller lifetime could be determined.

• The efficiency has been derived by a single loss calculation. Starting from the test conditions, the hydraulic efficiency was uprated to the full speed conditions. Volumetric losses and the friction losses of the impeller shrouds, bearings, and mechanical seals, etc., were taken into account. The casing surface has been protected with a coating, which protects the material and provides a smooth, low loss surface.

• The hydraulic shapes have been checked during the machining with templates to prevent any deviation of the characteristic and to meet the guarantee values in head of −0 to +2 percent head at rated flow.

• Thrust calculations based on extensive experience have been carried out to keep the residual thrust within the bearing capacity limits for new and worn conditions. In connection with this, an additional throttle sleeve has been installed on the nondriven-end to reduce the mechanical seal pressure.

• The lateral rotor analysis consisted of a damped lateral Eigen-analysis, a coupling sensitivity analysis, and unbalance response analyses for both design clearances "new" and "worn" at the rated speed of 1492 rpm. An additional unbalance response analysis for the test speed of 1043 rpm has been carried out. The Eigen-modes satisfy all the damping and separation requirements specified by API 610, Eighth Edition. No coupling unbalance sensitivity factor (SF) of the rotor system is greater than two. Therefore, no vibration problems due to mechanical unbalance at the coupling location are to be expected. Thus, the damped lateral vibration analysis indicates that the machine has design integrity with respect to lateral rotordynamic behavior and that no vibration problems are to be expected during normal operation.

• The hydraulic shape of the s + s pump has been modeled in 3D due to the complex geometry. The finite element analysis of the casing of this large two-stage back-to-back pump was performed first to ensure the tightness of the horizontally split casing flange. Figure 20 shows the contact pressure of the split flange. Additionally, the casing stresses and the bolt stresses were of interest as well as whether the seal cavity displacements were too large to ensure tightness at this position. Load cases analyzed were the pretension of the casing bolts, the operational load case, the design case, and the hydrostatic test. The gasket was simplified to linear behavior. The contact pressure is high enough between and around the bolt holes to guarantee tightness of the pump. The casing stresses caused by pressure loading are low in general. There are some stress peaks at the cutwater edges, but they will not exceed low cycle fatigue limits.

• The torsional analysis consisted of an Eigen-mode analysis, a lateral-torsion interaction response calculation, and a transient response analysis for both possible transient excitations, i.e., the switch on grid and the short circuit load cases. The result proves that the stresses due to the excitation have no adverse effect on the pump train. The presence of an autotransformer attenuates considerably the transient response to the switch to grid excitation; the maximum peak value of internal torsional stresses remains in the elastic range, and sufficient fatigue life of the train is assured.
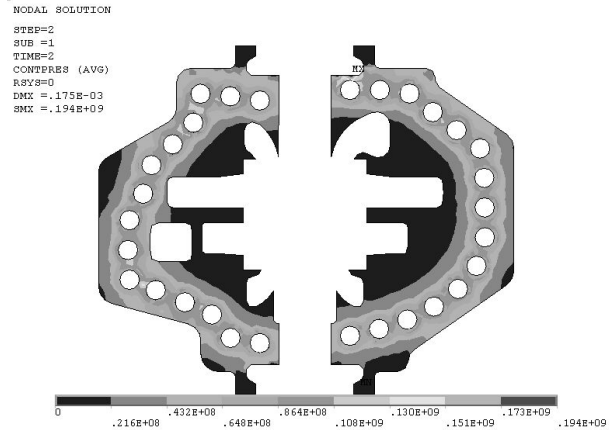

Figure 20. Contact Pressure of Horizontally Split Flange.

• Pump and motor are mounted on a strong, heavy concrete foundation block. The exciting forces from the hydraulic pulsation have to be damped by flexible Sylomer® damping strips to avoid excessive vibration velocities on the bearing housing.

• In addition, the mechanical seal behavior has been analyzed with finite element calculations to find the temperature rise and leakage losses.

All the above calculations were necessary, due to the fact that the delivery time was extremely short and the manufacturer wanted to assure that the shop and field tests would run smoothly without any unexpected problems.

PUMP TESTING

Model Tests

• For Case 2, the s + s pump has been built with an impeller size of 400 mm (15.75 inches). With this pump, the flow-head-power-NPSH-curve has been measured and also the pressure distribution at different positions in the volute, and the crossover has been measured to analyze the thrust balancing and the efficiency. With this model pump, the pump diameter could be scaled up to 815 mm (32.1 inches).

• For Case 3, the 3d pump has been built with an impeller size of 325 mm (12.8 inches). The client wanted to see the model test running in pumping, pump braking, and turbine mode, and to receive curves for the water hammer calculation. In addition, the flow-head-power-NPSH-curves have been measured for the nominal and reduced impeller size.

The time to manufacture and test a model pump has to be added to the project time. For Case 1, the test results of Case 2 s + s machine have been adopted to determine the hydraulic layout basic values for the full size of the prototype.

Most specifications ask that the manufacturer demonstrates the pump performance test at full speed and, if possible, with the client driver. This is a difficult task and would extend the delivery time and the testing cost, due to limited electric power supply on most test bed installations. In most cases, a calibrated large test bed motor is not available that can run for a short period at high power loading to cover the full performance range. In cases where a full load test is essential, the planning of such a test has to be started at the time when the order is placed. A full load test is very costly and cannot guarantee that all results are directly comparable to the plant. It is therefore preferable to perform a factory acceptance test at reduced speed. The test procedure should define exactly in which way the test results at reduced speed have to be converted to the full speed. There are hydraulic data and mechanical data to be tested, e.g., hydraulic data such as flowrate, head, power, NPSH, efficiency, and uprating to full speed must be defined. The test

layout should allow the measurement to be taken at the position of uniform flow distribution. Mechanical data such as shaft displacement, bearing housing vibration, bearing temperature, and noise measurement have to be defined for the test conditions.

Typical test results are shown for each case in tables wherein the values can be easily compared, i.e., guarantee value, shop test results, and field test results. The shop test results are usually of higher quality with respect to the measurement accuracy. In the field, there are limitations to the accuracy of the measurements. Onsite test thermodynamic efficiency measurements are possible; however, external mechanical losses must be considered. Onsite the flowrate usually cannot be adjusted, as on the test bed, and some operating points of the shop tests cannot be repeated. The NPSH test should not be carried out in the plant because of the risk of damage to the pump by cavitation. In the plant, the hydraulic performance and the mechanical values are of great interest.

Tables 4 through 7 list the test results for the main pumps for all three cases. For Case 1, all pumps have been tested in the shop and one in the field (refer to Table 4 for the results). For Case 2, all pumps have been tested in the shop and in the field as shown in Tables 5 and 6. For Case 3, all pumps have been tested in the shop and in the field as shown in Table 7.

Table 4. Test Results from Shop and Field Tests of Main Pump in Case 1.

Condition	Unit	Shop	Field	Guarantee
Speed	RPM	(1'052)	1'490	1'490
Flowrate	m³/h	(4'458)	6'314	6'314
Head	m	(243.9)	488.0	486.5 $^{-0+9.7}$
Efficiency	%	(89.1)	89.8	88.8
Power	kW	(1'645)	9'278	9'362
NPSH-3%	m	(10.4)	n.a.	25.1
DE-shaft-displacement	цm	(26/25.5)	36/34	70
NDE-shaft-displacement	цm	(21/24.5)	27/31	70

(…) Values in bracket belonging to the corresponding speed

Table 5. Test Results from Shop and Field Tests of the "S + S" Main Pump in Case 2.

Condition	Unit	Shop	Field	Guarantee
Speed	RPM	(1'238)	(1'630)	1'740
Flowrate	m³/h	(3'757)	(4'946)	5'280
Head	m	(254.5)	(441)	502
Efficiency	%	(88.8)	(89.3)	89.1
Power	kW	(2'934)	(6'656)	8'106
NPSH-3%	m	(13.2)	n.a.	28.0
DE-shaft-displacement	цm	(19.6/25.5)	(32/41)	70
NDE-shaft-displacement	цm	(18.6/13.7)	(33/39)	70

(…) Values in bracket belonging to the corresponding speed

Table 6. Test Results from Shop and Field Tests of the "1d" Main Pump in Case 2.

Condition	Unit	Shop	Field	Guarantee
Speed	RPM	(1'100)	(1'100)	1'170
Flowrate	m³/h	(14'892)	(14'892)	15'840
Head	m	(179.5)	(179.5)	212
Efficiency	%	(91.8)	(91.5)	90.9
Power	kW	(7'935)	(7'961)	10'061
NPSH-3%	m	(6.7)	n.a.	22.0
DE-shaft-displacement	μm	(24.5/22.5)	(41/40)	70
NDE-shaft-displacement	μm	(24.5/23.0)	(40/37)	70

(…) Values in bracket belonging to the corresponding speed

Table 7. Test Results from Shop and Field Tests of the "3d" Main Pump in Case 3.

Condition	Unit	Shop	Field	Guarantee
Speed	RPM	(437)	1'200	1'200
Flowrate	m³/h	(2'950)	8'100	8'100
Head	m	(47.7)	359	351.2
Efficiency	%	(89.0)	90.8	90.0
Power	kW	(431)	8'727	8'554
NPSH-3%	m	(1.7)	n.a.	15.0
DE-shaft-displacement	μm	(28/18)		80
NDE-shaft-displacement	μm	(37/21)		80

(…) Values in bracket belonging to the corresponding speed

CONCLUSION

The three examples represent a variety of water transport systems. It could be shown that the geodetic profile of the pipeline dictates the economical split of parallel pump units. The fixed speed selection can be taken into consideration when the intermediate storage capacity is sufficient and the system curve is relatively flat. In all other cases, the variable speed solution has an economical advantage to work over a wide operating range close to the best efficiency of the pump and to avoid any throttling of the generated head. In addition, the variable speed arrangement reduces the loading of the pumps especially during the starting phase of the plant.

In a system with fixed speed drivers, the pump flow runout point, which is the intersection of the single pump curve with the system characteristic, has to be checked for safe operation without cavitation damage for the specified operating period.

The approach of designing special water transport pumps as described above is essential to reduce the risk of unexpected failure during commissioning and operating. A further risk in large water transport plants is the long-term storage treatment of the equipment between the shop tests and the commissioning phase. When pumps are left in the field without special precaution, extended checks on the pumps have to be carried out prior to the commissioning phase. With all above measures, the risk of failure can be reduced and the pump can operate properly as expected in the transport system.

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