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REPORT

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LITERATURE REVIEW ON THEORETICAL PUMP AND MOTOR EFFICIENCY OF SUBMERSIBLE PUMP SYSTEMS

Project acronym: OptiWells-1

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Literature review on theoretical pump and motor efficiency of submersible pump systems

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Chapter 1
Introduction

1.1 Objectives and scope

1.1.1 Background and objectives of the document

A German survey of operating submersible water pumps has shown that over 80% of the pumps were operating at global efficiencies ranging from 36% to 48% (Teders 2008). On the other hand, in a case-specific study in Northern Germany, groundwater well pumps were operating at efficiencies as low as 18% and 21% (Plückers 2009). This shows that the further improvement in energy efficiency of water abstraction system is feasible.

The purpose of this document is to provide an introduction to energy-efficient pumps and motors, and to review theoretical considerations on the potentials for high-efficiency pump and motor systems. The system “pump + motor + drive” will be referred as to the “pump system” given the embedded nature of submersible pump motors in the pump’s casing.

After introducing the role of pumps in the water abstraction system (Chapter 1), this report describes submersible pump and motor technology (Chapter 2) and explores ways of optimizing the energy demand of pumping systems used for groundwater abstraction (Chapter 3).

This document shall provide a sound basis to evaluate the potential enhancements that can be implemented on pumping systems. It shall assist the identification of solutions for an energy-efficient groundwater pumping.

→ *More information on the available innovative pump technologies will be provided by the OptiWells-1-deliverable D 2.2 (International market review of pumps available for groundwater wells), which is complementary to the present deliverable D 2.1.*

1.1.2 Scope of the document

The project OptiWells focuses on drinking water abstraction, defined as the domain commencing with the entry of water into water intake systems (i.e., here drinking water wells), and ending with the entrance in the water works (purification plant or distribution station). The scope of this document is narrowed to the pumps dedicated to groundwater abstraction (Fig. 1).

The most utilized pumps for water abstraction in Europe are electrical submersible pumps, which are centrifugal pumps particularly fitted for the use in wells. This study focuses on this type of pumps, and shall not be extrapolated for other pump types or similar pumps dedicated to other uses.

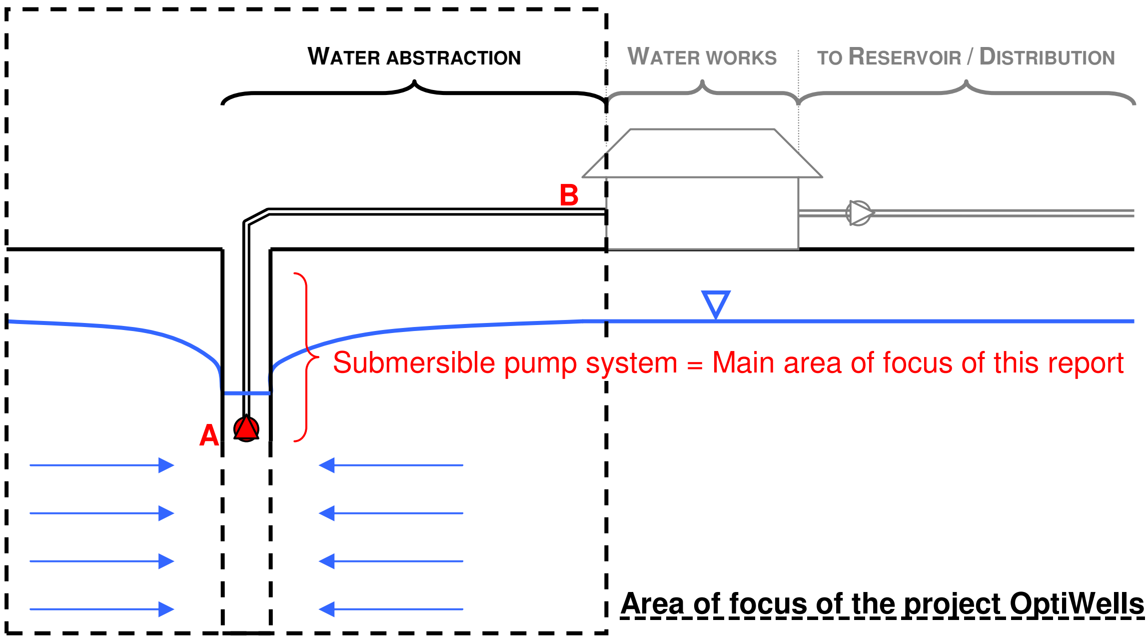


Fig. 1: schematic of groundwater abstraction, purification and distribution.

1.2 Hydraulic head losses in pipe and pump hydraulics

1.2.1 Hydraulic head

The Bernoulli equation states that the hydraulic head at a point A, H_A (m) is given by:

$$H_A = z_A + \frac{p_A}{\rho \times g} + \frac{V_A^{\;2}}{2 \times g}$$

with z_A the geometrical elevation (m), p_A the pressure (Bar) and V_A the flow speed (m/s). In general, for well-designed piping systems, the term $V_A^2/(2\text{x}g)$ can be neglected in comparison to the other terms. For instance, for a typical flow velocity of 1m/s, it equals 0.05m (i.e. 5 cm), while the other terms are generally greater than 10m. Hence,

$$H_A \approx z_A + \frac{p_A}{\rho \times g}$$

On highly encrusted, aged pipes, the diameter and section of the pipe may be dramatically reduced, resulting in a much larger $V_A^2/(2\text{x}g)$ term, and then this approximation is no longer possible.

1.2.2 Head losses

While flow velocity (or, implicitly, flow rate Q) is generally not used for calculating the hydraulic head, it is however the critical parameter for the determination of head losses. When flowing from a point A to B, while the total energy is conserved, parts of it are "dissipated" (i.e. no longer available for the fluid) due to friction resulting mainly in thermal and acoustic dissipations due to local turbulence. These dissipations are called head losses. The head conservation equation is written as:

$$H_A = H_B + j_{A \to B}$$

with $j_{A \to B}$ the hydraulic head losses (m) between point A and B. In the direction of flow, the head is decreasing and $H_B < H_A$ ($j_{A \to B} > 0$). Friction is caused by "obstacles" to the water flow, such as:

- pipe roughness along the pipe network, caused by imperfections on the inner pipe surface, resulting in *major* head losses;
- valves, bends, fittings, or other discontinuities on the pipe network, resulting in *minor* head losses.

At the scale of a pipe network, minor head losses usually constitute around 10% of total losses (Mosé and Roche 2005). Head losses are considered as a function of the square of flow velocity. For major head losses, the most common expression is the Darcy-Weisbach equation:

$$j_M = \lambda \times \frac{L}{D} \times \frac{V^2}{2 \times g}$$

where L is the pipe length, and D its inner diameter. This relationship is sometimes written as (Haakh 2009):

$$j_M = \alpha \times Q^2$$

with

$$\alpha = \frac{8 \times L \times \lambda}{\pi^2 \times g \times D^5}$$

The factor λ is the Darcy friction factor and is dependent upon the flow Reynolds number (Re), the roughness (k) and inner diameter (D) of the pipe. It can be estimated by the Colebrook equation or using the Moody-Stanton chart (Appendix). Other expressions of linear head losses also exist, but are used more sparsely or only locally (for instance, Lechapt and Calmon in France, Hazen and Williams mostly in North America).

Minor head losses are calculated by:

$$j_m = \zeta \times \frac{V^2}{2 \times g}$$

with ζ the minor loss coefficient. Minor loss coefficients can range from 0.01 to 1.50 or higher, depending among others on the type of obstacle, angle of bend and roughness of the pipe (Mutschmann and Stimmelmayr 2007). For relatively short pipe systems with a large number of bends and fittings or in the case of throttled valve, minor losses can easily exceed major losses.

1.3 Role of pumps in water abstraction

1.3.1 Role of pumps in a hydraulic system

Basically, a pump can be defined as a machine that imparts energy to a fluid by increasing hydraulic head (Sterrett 2007). In the water abstraction system, pumps are used to convey water, to overcome the geometrical height difference between the groundwater table and the water works, and since water flow in pipes always generates head losses, they also need to overcome these.

1.3.2 Requirements and calculation of minimum total dynamic head

Pumps are required for water abstraction to convey the water from the well to the water works. The calculation of the total dynamic head[1] of the pump H_n bases on:

[1] also known as “lifting height” or “manometric height”.

- the difference in height Δz between the static water level in the well and the water level in the water works (where water can be either at atmospheric pressure – free surface – or under hydraulic pressure),
- the hydraulic head losses j in the discharge pipe of the pump,
- the drawdown in the well s created by the groundwater abstraction, which adds to the static geometrical elevation. The drawdown corresponds to the difference between static and dynamic water level.

In fact, H_n is the sum of the positive difference in geometrical height, the losses and of the difference in pressure between point A (well) and point B (at the water works):

$$H_n = \underbrace{(z_B - z_A)}_{\Delta z + s} + \underbrace{j_m + j_M}_{j} + \underbrace{\frac{(p_B - p_A)}{\rho \times g}}_{\Delta p}$$

Fig. 2 shows schematically the hydraulic head at different points of the system. Submersible pumps are located beneath the well water level. Depending on the installation, the remaining relative pressure at the water works p_B can be zero (free surface flow at water works entrance), e.g. if the water is re-pumped for purification processes in the water works.

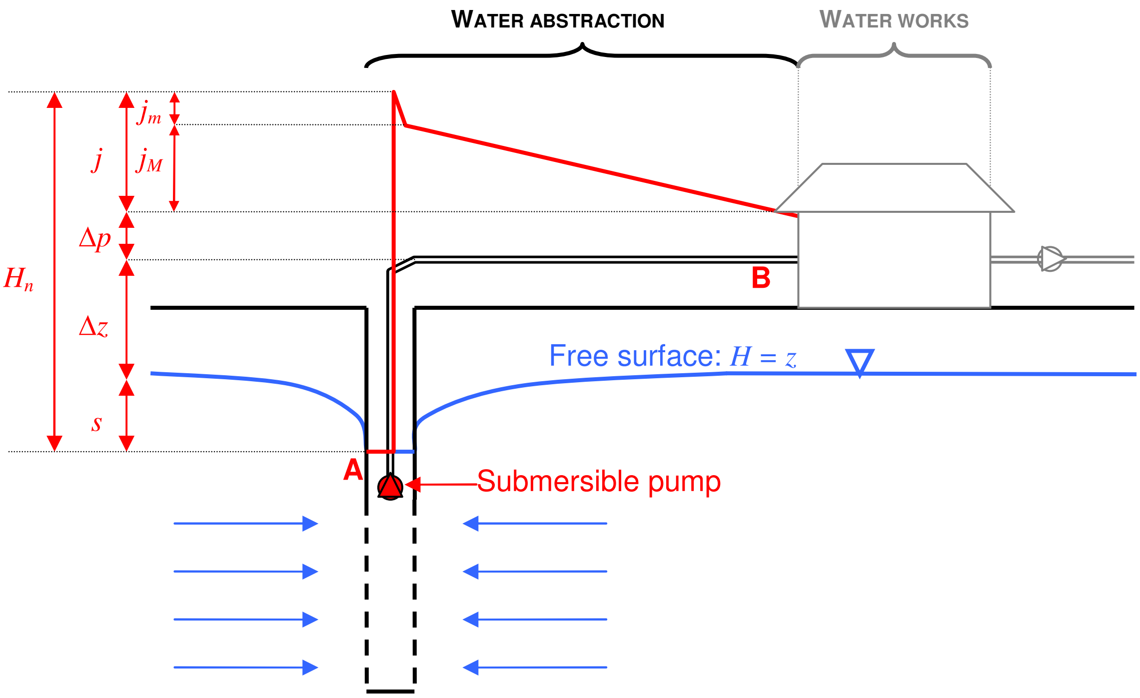


Fig. 2: hydraulic head in the water abstraction system (not to scale – the head losses are only given for illustration).

Chapter 2
Description of submersible pump systems and considerations on efficiency

2.1 Pump and motor description

2.1.1 Pump description

In drinking water applications, 99% of utilized pumps are centrifugal rotodynamic pumps (Mosé and Roche 2005). As stated in Section 1.2, this report focuses on electrical submersible pumps, which are small-diameter centrifugal pumps that fit in wells and enable to minimize well construction costs (Sterrett 2007). They comprise (Fig. 3):

- a pump intake chamber, protected by a screen or grid;
- a series of impellers linked to a rotating shaft, imparting energy to the fluid via several vanes or blades; several impeller stages can be stacked;
- a series of pump bowl casings, which is the stationary element directing the fluid flow via diffusers;
- a check valve, preventing back flow when pump is stopped;
- a discharge pipe (or drop, rise pipe), which is directly connected to the head of the well. The discharge pipe also acts as a hanger for the pump

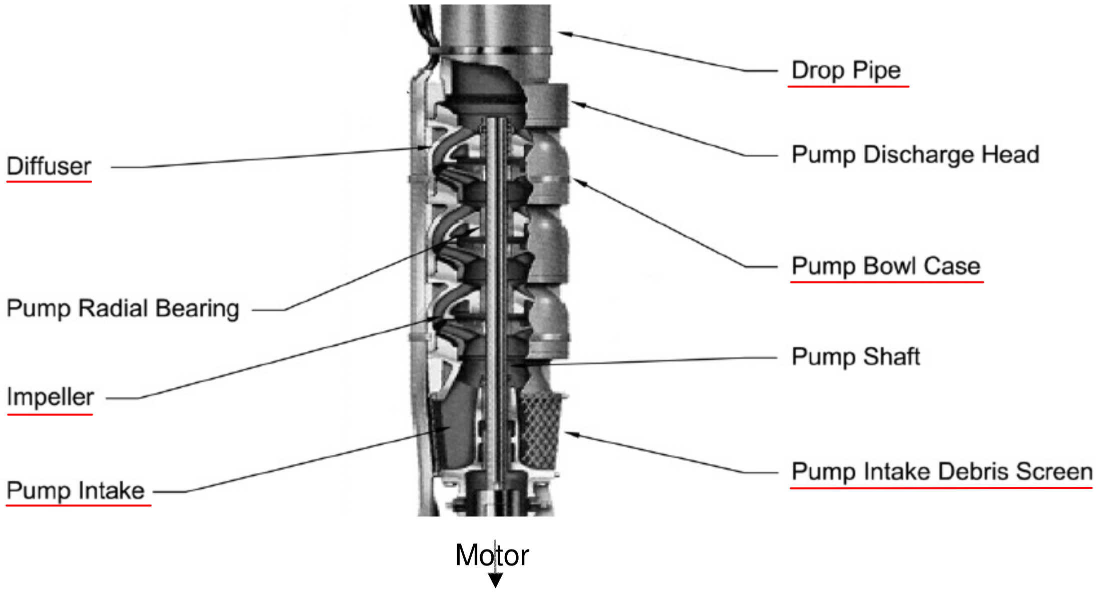


Fig. 3: typical submersible pump (Sterrett 2007).

Bearings are also an essential component of pumps and motors, and enable (almost) friction-free constrained rotation of the shaft and suspension of the rotor.

Often, submersible pumps are multi-staged. This association of several stages works in a similar way to pumps in series, and enable to reach higher lifting heights for small impeller diameters (Mutschmann and Stimmelmayr 2007) (see also 2.1.4 for the association of pumps). Fig. 3 shows a three-stage submersible pump with radial impeller.

Impellers of centrifugal pumps can be either radial (for low-flow/specific speed, high head), half-radial (mixed conditions) or axial (for high-flow/specific speed, low head). Impellers used in submersible pumps are generally radial or sometimes half-radial (mixed-flow) to reach important lifting heights (Fig. 4).

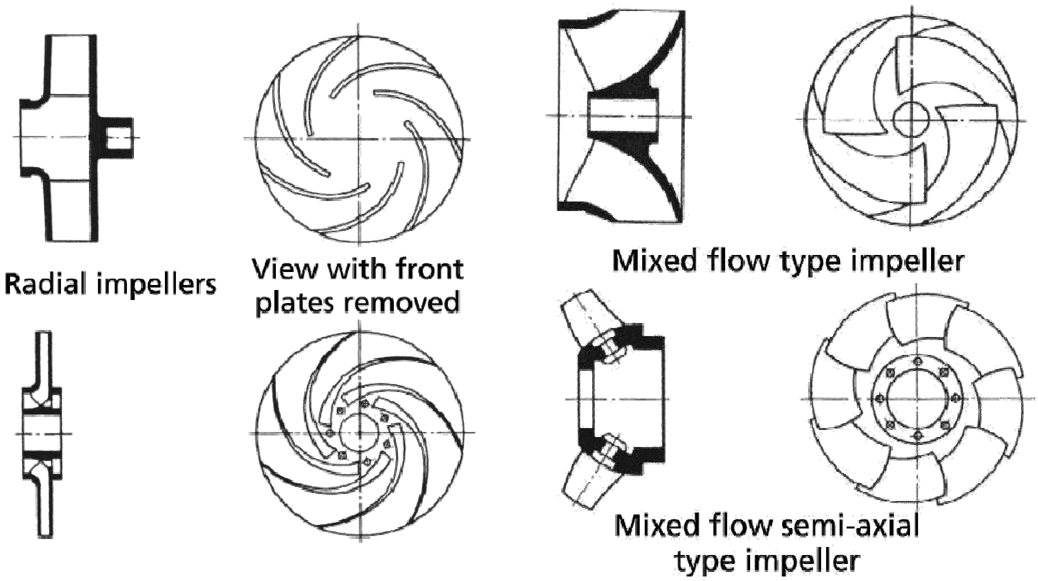


Fig. 4: radial and half-radial pump impellers (http://www.pumpfundamentals.com/impeller.htm).

The discharge is roughly directed at a right angle to the impeller centre and pump shaft axis (Sterrett 2007). The design and direction of the blades is related to an angle β that influences the pump curve (Fig. 5) (Mosé and Roche 2005).

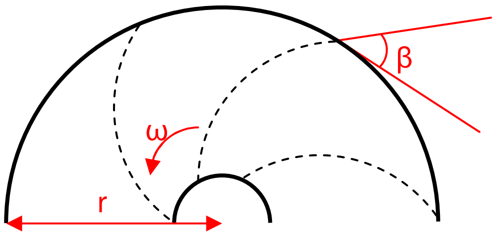


Fig. 5: pump impeller characteristics, adapted from (Gülich 2010).

Typically, electrical submersible pumps include a check valve on the discharge side of the pump (i.e. above). Its purpose is to keep the discharge pipe full of water, and prevent the pump and motor from rotating in the reverse direction, which could damage the motor, especially if the pump is given a new start command during the backspin (Sterrett 2007).

2.1.2 Motor description

The motor is connected to the impeller via their respective shafts (Fig. 3 and Fig. 6). The connection between shafts is made by a close mechanical coupling. The motor, generally an alternative current (AC) electrical induction motor[2] is located beneath the pump and comprises (Fig. 6):

- a watertight motor stator (stationary element);
- a motor rotor, which is connected to the rotating motor shaft,
- a pressure and volume equalization or compensation chamber, which is equalizing the differences in pressure of the well water (usually a diaphragm).

[2] for high power systems, sometimes synchronous motor with a wired rotor are used.

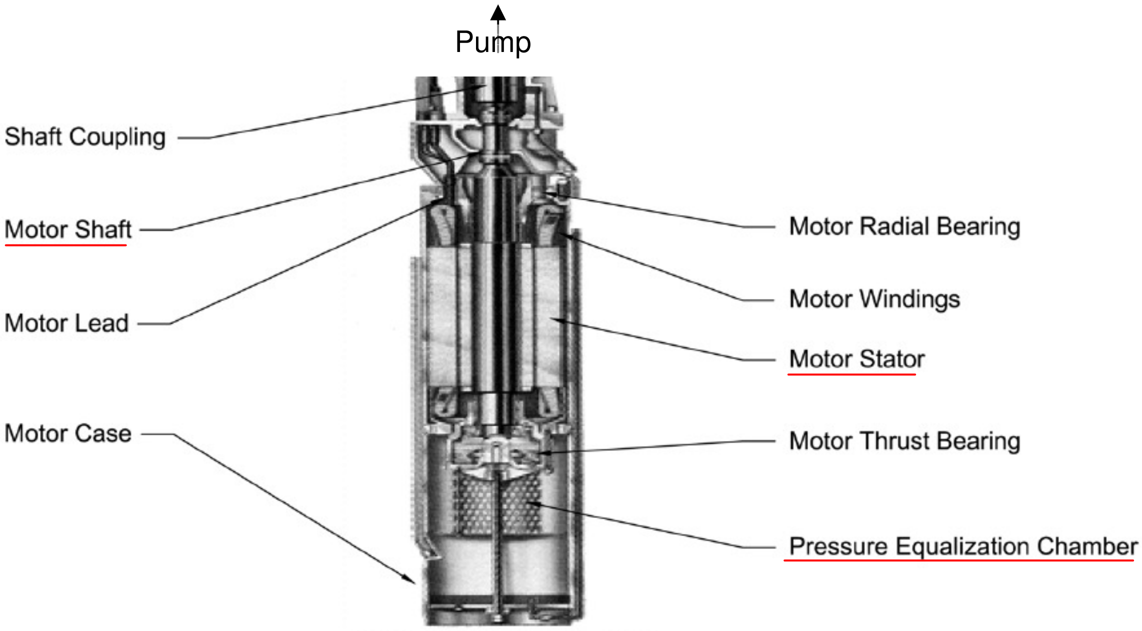


Fig. 6: typical motor of a submersible pump (Sterrett 2007).

For submersible pumps, the motor and the pump, although they are two separate elements, are located in a unique casing. This is a major difference to other centrifugal pumps used, e.g., for water distribution, which use aboveground motors.

The AC induction motor consists of two assemblies, stator and rotor (Fig. 7). The stator structure is composed of steel laminations shaped to form poles. Copper wire coils are wound around these poles. These primary windings are connected to a voltage source to produce a rotating magnetic field. Three-phase motors with windings spaced 120 degrees apart are standard. The denomination "squirrel-cage motor" is often used since the arrangement of the rotor bars resembles a squirrel cage (Mutschmann and Stimmelmayr 2007). The area between the stator and rotor is filled with a solution that is generally mostly water mixed with other additives. The stator itself is also hermetically sealed to avoid penetration of water (FRANKLIN ELECTRIC 2007).

Because of its submersible nature, the pump and motor system is directly cooled down by the water of the well before entering the suction screen of the pump. Water cooling is critical to the longevity of the motor insulation as well as the lubrication of the motor thrust bearing (Sterrett 2007). In some situations, to enable sufficient cooling, the water-flow velocity past the motor can be increased using cooling shrouds to force the pumped water past the surface of the motor, especially when the pump is located directly close to the screen, with the flow entering the pump not passing along the motor.

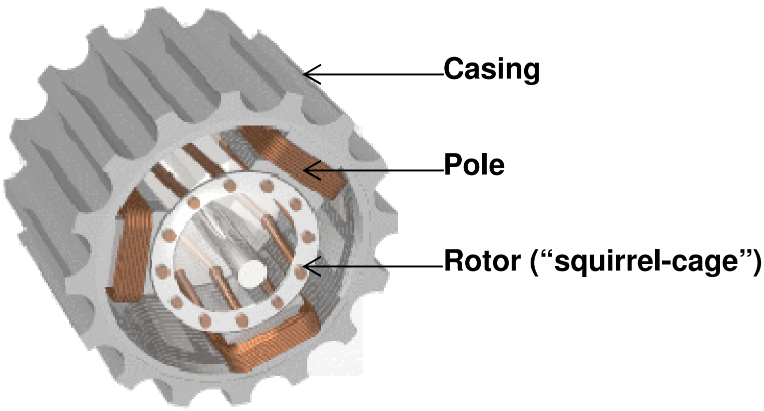


Fig. 7: three-phase squirrel-cage induction motor (http://en.wikipedia.org/wiki/Induction_motor).

Motor load is defined as the ratio between actual electrical absorbed power, P_{elec}, and the nominal rated electrical power of the motor, P_{nom} (Kaya, Yagmur et al. 2008):

$$Load = \frac{P_{elec}}{P_{nom}}$$

Load is a critical factor for pump efficiency (Section 2.2.2).

The synchronous rotation speed of an induction motor N_s (rpm; ignoring slip, see below) depends on the supply frequency f (Hz) and on the number of magnetic poles n_{poles} (BPMA 2002):

$$N_s = \frac{120 \times f}{n_{poles}}$$

As such, the speed for a given motor is normally fixed since both the supply frequency and the number of poles are fixed. The majority of groundwater pump motors are equipped with 2 or 4-pole motors. For a 50 Hz supply and a 2-pole motor, the speed of the induction motor is 3000 revolutions per minute (rpm).

In the motor, the interaction of currents flowing in the rotor bars and the stator's rotating magnetic field generates a torque. The rotor speed always lags the magnetic field's speed, allowing the rotor bars to cut magnetic lines of force and produce useful torque τ ($\tau = P/\omega$). This relative speed difference is called the slip and calculated by:

$$s = \frac{N_s - N}{N_s}$$

where s is the slip, N_s the synchronous speed of magnetic field (rpm) and N the shaft rotating speed (rpm). The slip increases with load and decreases with increasing rated power (Table 1).

The typical motors used for submersible pumps have rated powers of 0.3-150 kW and above (ITT 2011). Usually, a 3% slip is taken for a rough estimate, and thus a 1500 or 3000 rpm motor (synchronous speed) respectively corresponds to an effective 1450 or 2900 rpm shaft speed, called the rated speed or nominal speed.

Table 1: motor power range and typical slip for squirrel-cage induction motors (http://www.engineeringtoolbox.com/electrical-motor-slip-d_652.html and (Mutschmann and Stimmelmayr 2007)).

Power (kW)	0.35	3.5	10	35	175
Typical slip s (-)	5%	3%	2.5%	1.7%	0.8%

2.1.3 Pump curve

The pump curve[3] is the curve giving the Total Dynamic Head (TDH) H_n (m water column) as a function of the discharge Q (m^3/s). The *theoretical TDH, H_l* (m), is given by (Gülich 2010):

$$H_l(Q) = a + b \times Q$$

where the "constants" a and b are of course dependent of pump design and operation. As seen in Section 1.2, the change in fluid direction in the pump and shocks with the impellers induce head losses, which are of the form:

[3] also known as "pump characteristic curve", "performance curve" or "head-capacity curve".

$$j_{pump}(Q) = k \times \frac{V^2}{2 \times g}$$

The actual TDH H_n (m) is the difference between theoretical TDH and head losses:

$$H_n(Q) = H_t(Q) - j_{pump}(Q)$$

Pump curves are decisive for the selection of the most appropriate equipment. In general, submersible pump curves are designed steeper than other centrifugal pumps, i.e. high variations in lifting heights result in relatively small variations in discharge. They are designed so to prevent high flow rate variations when the dynamic water level varies, e.g. due to seasonal variations. A typical example of pump curve for a centrifugal submersible pump is given in Fig. 8.

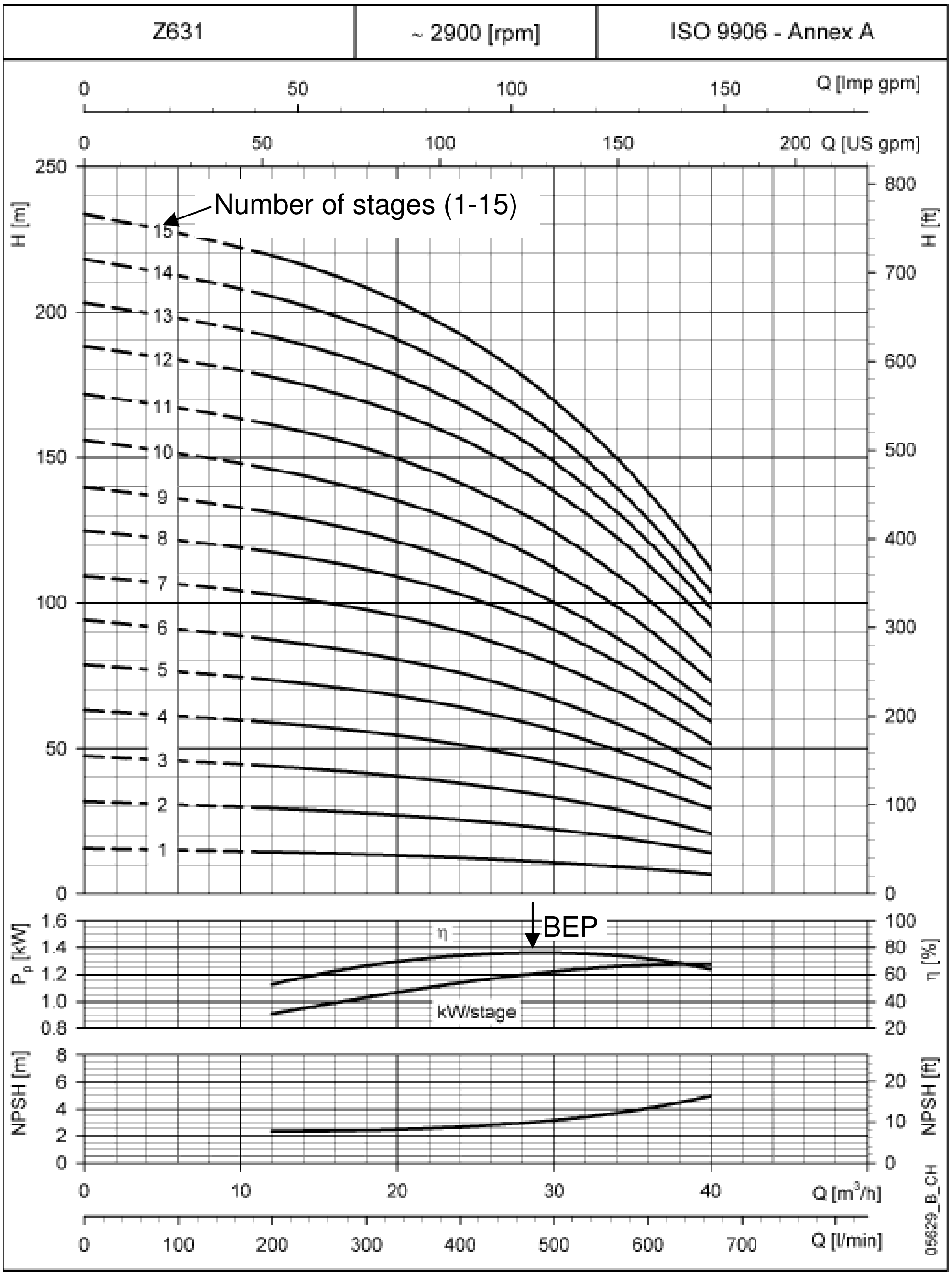


Fig. 8: typical pump curve for a 6" centrifugal submersible pump, the BEP is here 78.5%. For this type of pump, configurations of 1 to 15 stages are available (ITT 2011).

2.1.4 Pump system operation point

A pump is operating within a hydraulic system, and as such, its operating point[4] depends upon the system (network) curve. In groundwater wells, due to pumping, the groundwater table varies with time. Hence, three characteristic (H,Q) curves are necessary to determine the pump operating point:

- the pump curve (TDH-discharge – green bold curve on Fig. 9);
- the pipe curve (head losses-discharge – red dotted line on Fig. 9);
- the groundwater and well drawdown curve (drawdown-discharge – dark blue dotted line on Fig. 9).

As stated earlier (Section 1.3.2), the total dynamic head of the pump H_n shall cover the sum of the geometrical elevation Δz, the well drawdown s and the head losses j, thus the pump operating point is determined for the discharge Q_{op} by:

$$H_n(Q_{op}) = \Delta z + s(Q_{op}) + j(Q_{op})$$

Fig. 9 shows this equation graphically, the sum of $\Delta z + s(Q_{op}) + j(Q_{op})$ is indicated in dark blue (required head).

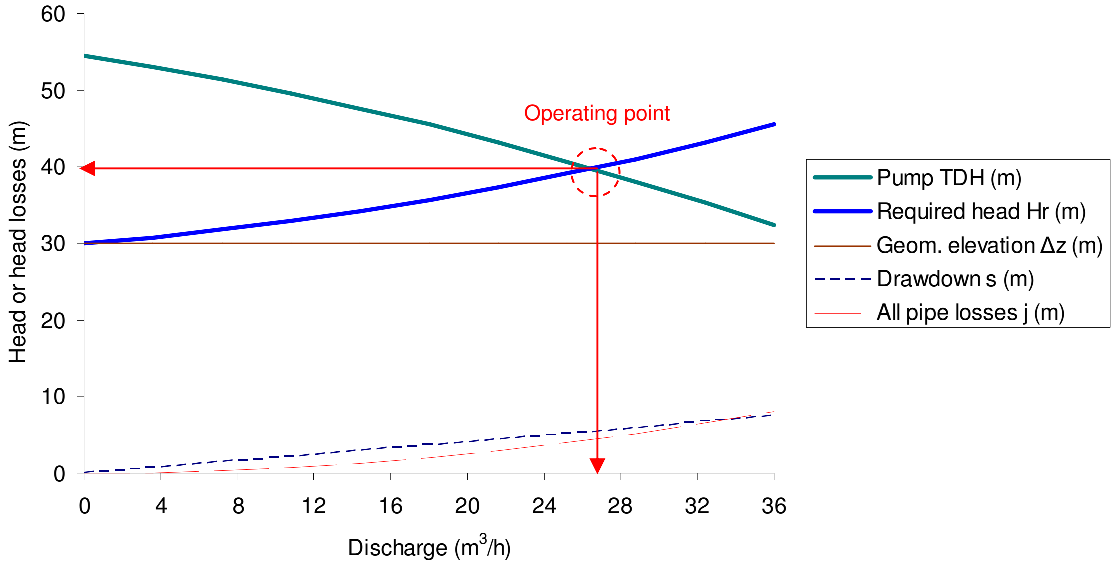


Fig. 9: theoretical example of how the pump operating point can be determined. Here, $Q_{op} \approx 27$ m^3/h (fictive example).

To achieve higher water delivery volumes or a higher water head or pressure, provided a sufficient supply from the well field, it is possible to operate pumps in parallel or in serial:

- when pumps are arranged in *parallel*, their resulting pump curve is obtained by adding their flow rates at the same head (bold red curve on Fig. 10). Pumps are arranged in parallel to overcome larger volume flows than one pump can handle alone;
- when pumps are arranged in *serial*, their resulting pump curve is obtained by adding their heads at same flow rate (bold dotted orange curve on Fig. 10). Pumps are arranged in serial to overcome larger elevations or system head loss than one pump can handle alone.

[4] also known as "duty point".

The new operating point of the system is defined by using the pump association's pump curve and the system curve. This means for instance that using two similar pumps in parallel does not mean that the flow is multiplied by two, since the system curve will influence the new operating point.

A particular case of association in serial is the multi-stage submersible pump, which consist of several impellers arranged in serial, thus multiplying the head capacity of the pump, while keeping the same overall discharge (Fig. 3 showed a three-stage pump).

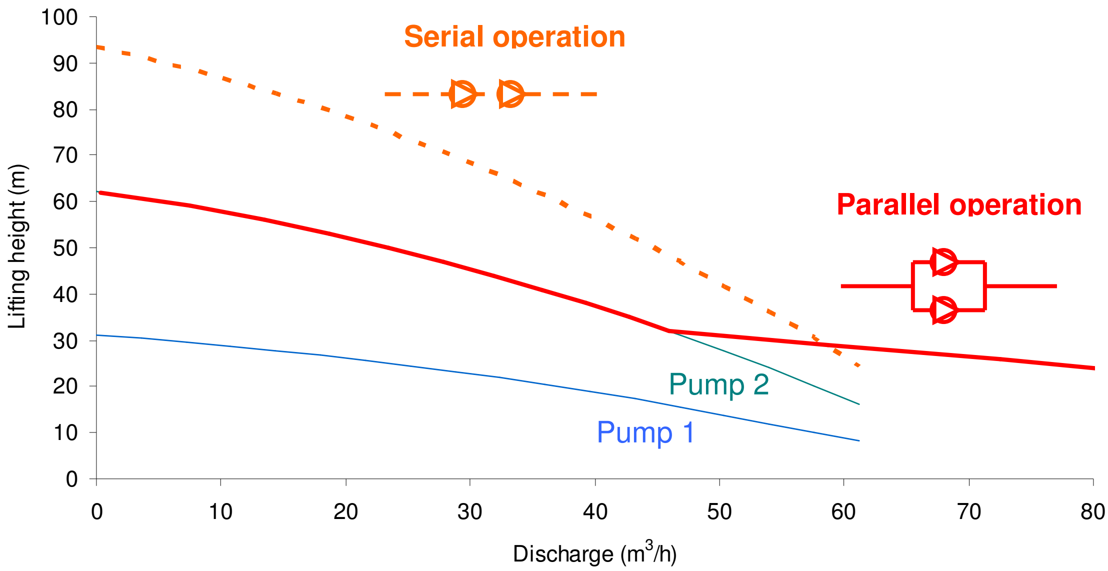


Fig. 10: association of pumps in parallel and in serial (fictive example for two pumps).

The Best-Efficiency Point (BEP) is the "most efficient operating point". It is the flow rate at which the pump operates with the lowest specific energy (volume moved per energy unit) (Berry 2007). The pump efficiency decreases when getting away from the BEP, on the right or left-hand sides of the curve (Fig. 8).

2.1.5 Important parameters for pump system selection and sizing

Following factors are to be considered when selecting submersible pump equipment (Mutschmann and Stimmelmayr 2007):

- required operating point (H,Q), as well as expected minimum and maximum discharge, minimum and maximum drawdown and geometrical elevations in order to determine the Net Positive Suction Head (NPSH);
- type of well, diameter, casing characteristics;
- chemical water characteristics, especially corrosive and/or scaling potential, dissolved oxygen, iron, chloride, manganese;
- switching: additional start-up transformator, soft start-up, star-delta start-up, and nature of start-up command (manual, remote, sensor-controlled);
- type of electricity supply: supply voltage and frequency.

The pump curve is used to size the appropriate pump system. Often, the worst-case scenario for the pumping rate is used (high demand) to determine the worst-case operating point of the pump and the power capacity of the motor. A security margin is also often included, resulting in motor capacities exceeding 10-15% of the required capacity (Mutschmann and Stimmelmayr 2007).

2.2 Pump and motor efficiency

2.2.1 Expression of pump energy demand

The electrical power absorbed by a pump P_{pump} (kW) can be calculated by:

$$P_{pump} = \frac{\rho \times g \times Q \times H_{pump}}{\eta_{pump}}$$

with ρ the density of water (kg/m^3), g the gravity acceleration (m/s^2), Q in m^3/s and H_{pump} in m. η_{global} is the global pump efficiency (electricity-to-water). In other units (Q in m^3/h) and under standard conditions, this relationship can be rewritten as follows:

$$P_{pump} = \frac{Q \times H_{pump}}{367 \times \eta_{pump}}$$

Accordingly, the energy demand E_{pump} (kWh) for a given pump system operation during ΔT (hours) can be evaluated by:

$$E_{pump} = \frac{Q \times H_{pump}}{367 \times \eta_{pump}} \times \Delta T$$

2.2.2 Expression of motor energy demand

The electrical power absorbed by a triphase motor P_{motor} (kW) can be expressed by:

$$P_{motor} = \underbrace{\sqrt{3} \times U \times I}_{P_{app}} \times \underbrace{\cos\varphi}_{PF}$$

With U the voltage, I the current intensity and $\cos\varphi$ the power factor[5] (PF). Motor type and load influence the power factor, which is given by the pump manufacturers. Usually, pumps have relatively high power factors, ranging from 0.73 or above at half-load to 0.9 or above at full-load (Bauer 2005; ITT 2011).

Accordingly, the energy demand E_{motor} (kWh) for a pump driving motor in operation for ΔT (hours) can be calculated by:

$$E_{motor} = \sqrt{3} \times U \times I \times \cos\varphi \times \Delta T$$

In general, motors are oversized by 10-15% in order to guarantee a safety margin for cooling and to better withstand the high intensities at start.

2.2.3 Global efficiency and specific energy demand

The global pump system efficiency[6] (η_{global}) is the "electricity-to-water" efficiency, and can be written as the product of the pump (η_{pump}), the motor (η_{motor}) and the drive (η_{drive}) efficiency (Hovstadius, Tutterow et al. 2005; Mosé and Roche 2005; dena 2010):

$$\eta_{global} = \frac{P_{fluid}}{P_{elec}} = \underbrace{\frac{P_{fluid}}{P_{pump}}}_{\eta_{pump}} \times \underbrace{\frac{P_{pump}}{P_{motor}}}_{\eta_{motor}} \times \underbrace{\frac{P_{motor}}{P_{elec}}}_{\eta_{drive}}$$

[5] ratio between active electrical power P_{motor} (usable for the pump) and apparent power P_{app} (delivered by the mains, sum of active and reactive power) due to a lag between tension and intensity created by the power receiving device

[6] also sometimes known as the Coefficient of Performance (COP).

The absorbed electrical power by the pump system (pump + motor + drive) equals:

$$P_{elec} = \frac{P_{fluid}}{\eta_{global}} = \frac{\rho \times g \times Q \times H_{pump}}{\eta_{global}}$$

and thus, for Q in m^3/h and ΔT in hours:

$$E_{elec} = \frac{Q \times H_{pump}}{367 \times \eta_{global}} \times \Delta T$$

Often, the global efficiency is related to the specific energy demand of the pump system E_{spec} expressed in Wh/m^3/m by:

$$E_{spec} = \frac{E_{elec}}{(Q \times \Delta T) \times H_{pump}} = \frac{2.725}{\eta_{global}}$$

The specific energy demand of the pump system is a normalized parameter which enables to compare different pumps independently of their TDH or discharge rate, and it is directly related to the global efficiency (100% efficiency = 2.725 Wh/m^3/m).

Motor and pump efficiencies are decisive for the global efficiency (dena 2010). All these efficiency ratios may change with time and ageing of the mechanical or electronic equipment. It shall be noted that all these efficiencies are time-dependent, and that, in general, they decrease with time. The efficiencies given by the manufacturers are initial efficiencies and do not include the effects induced by operation.

Fig. 12 gives an overview of the global efficiencies in existing submersible pumping systems. Another German study by (Plath and Wichmann 2008) yielded similar results. These studies give a “snapshot” of current operating pump efficiencies, and not potential efficiencies of new equipments.

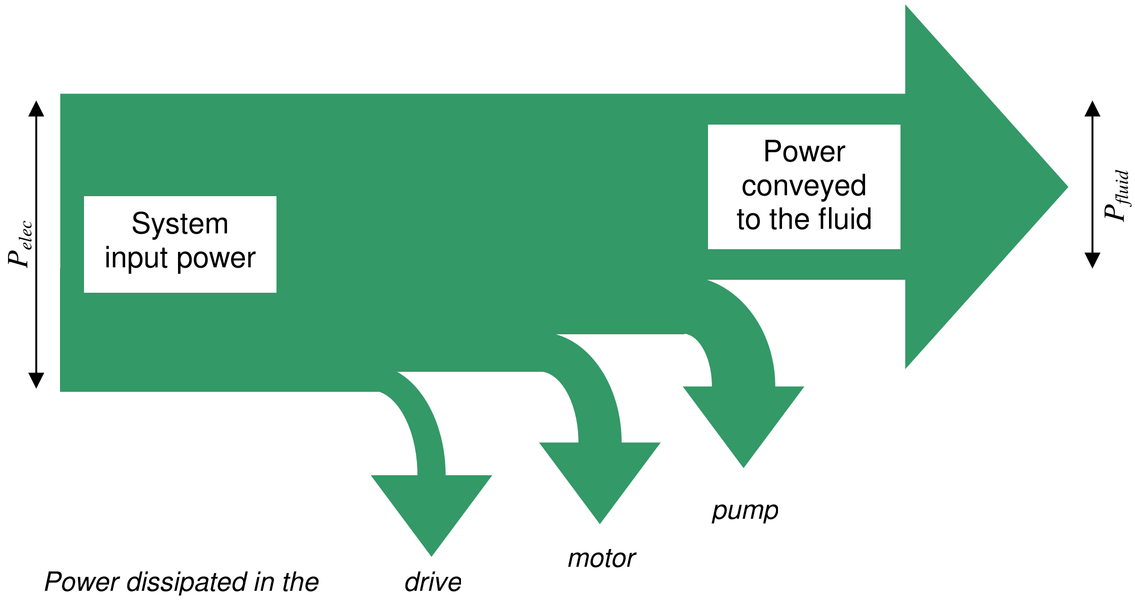


Fig. 11: visualization of global pump system efficiency (adapted from (Gülich 2010); not to scale).

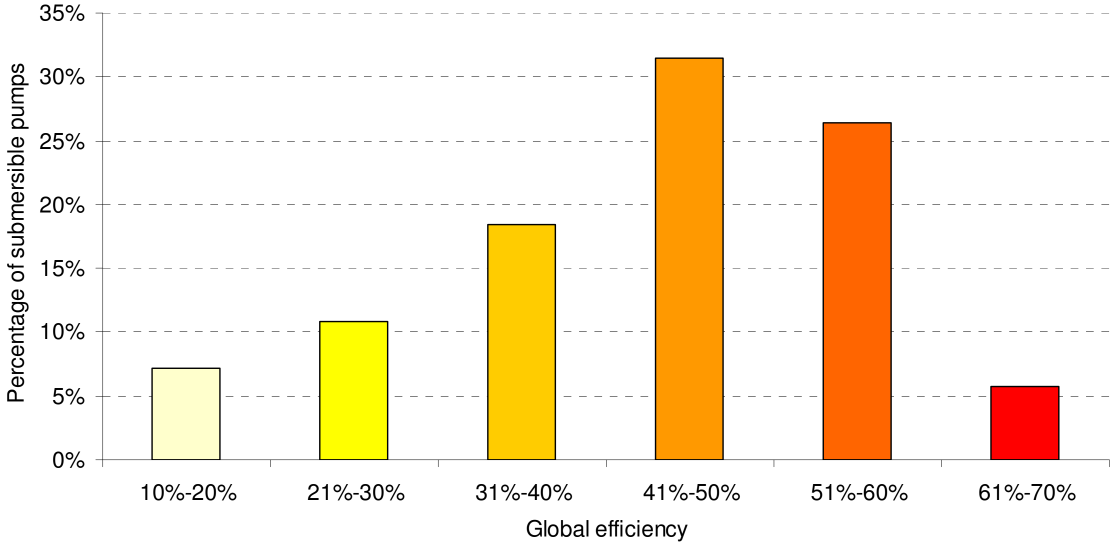


Fig. 12: ranges of global efficiency for submersible water pumps in operation in Germany - after (Boldt 2010). Note that this study also encompasses aged pumps, which explains the comparably low global efficiencies.

2.2.4 Pump efficiency

The pump efficiency is usually given by the manufacturer on the pump curve. It can be rewritten as the product of three "efficiencies":

$$\eta_{pump} = \frac{P_{fluid}}{P_{pump}} = \underbrace{\frac{Q_{fluid}}{Q_{pump}}}_{\eta_{volum}} \times \underbrace{\frac{H_n}{H_l}}_{\eta_{mano}} \times \underbrace{\frac{P_l}{P_{shaft}}}_{\eta_{mech}}$$

with Q_{fluid} and Q_{pump} the effective discharge of the fluid in the discharge pipe and the discharge of the pump (i.e. adding the recirculated discharge by-passing the impeller), and P_l the power used to reach the theoretical lifting elevation (without head losses). The pump power is the input power of the pump (i.e. conveyed by the shaft).

- The pump volumetric efficiency strongly depends on the clearance of sealing gaps (Gudbjerg and Andersen 2005). For a pump without internal leaks and for water (nearly incompressible fluid), this ratio is assumed equal to 100%. However, at low loads, the operation of the pump may induce additional hydraulic losses due to the partial recirculation of the fluid, i.e. when $Q_{fluid} < Q_{pump}$. Around the BEP, there are no recirculation losses (Gülich 2010);
- The pump manometric efficiency is impacted by hydraulic losses within the pump due to shocks and fluid friction in the system's volume, and are typically in the order of magnitude of 15% of input shaft power for centrifugal pumps (Gudbjerg and Andersen 2005), but can be higher in the case of submersible pumps due to the smaller impeller diameter;
- The pump mechanical efficiency is impacted by mechanical losses that generally occur in bearings and account for 1-5% overall losses in efficiency (BPMA 2002).

Pump manufacturers in general make no distinction between the different above mentioned efficiencies, which are integrated in the pump efficiency curve. As described earlier, the Best-Efficiency Point (BEP) characterizes the highest achievable pump efficiency for a given set of parameters (H, Q, impeller diameter and drive speed).

Typically, the larger the diameter of the impeller channels, the higher the pump efficiency, since the friction surface to impeller volume ratio is lower (Sterrett 2007). Also, submersible pumps usually rely on radial or mixed-flow impellers, which are, in general, less efficient (by 2-5%) than axial impellers due to more important turbulent flows induced by the sharper bending of the upper impeller plate (Gülich 2010).

Both these design restrictions explain at least partially why submersible pumps reach lower global efficiencies than larger centrifugal pumps: around 45-73% (Mutschmann and Stimmelmayr 2007) compared to centrifugal pumps which can reach hydraulic efficiencies of up to 80-90%. The pump efficiency is also affected by ageing, which lowers the efficiency over time.

2.2.5 Motor efficiency

The motor efficiency can be defined as the percentage of active input power from the drive that is converted to usable shaft power (Cornell Pump Company 2007). Motor losses are of two types: fixed losses - independent of motor load, and variable losses - dependent on load:

- fixed losses mainly consist of magnetic core losses (also called iron losses) induced by eddy current and hysteresis losses in the stator, and vary with the core material and geometry (and with input voltage);
- variable losses consist of resistance losses, miscellaneous stray losses and friction losses. Resistance to current flow in the stator and rotor result in heat generation that is proportional to the resistance of the material r and the square of the current i ($r \times i^2$ – Joule losses). Stray losses arise from a variety of sources and are difficult either to measure directly or to calculate, but are generally proportional to the square of the rotor current. Friction losses are caused by friction in the motor bearings.

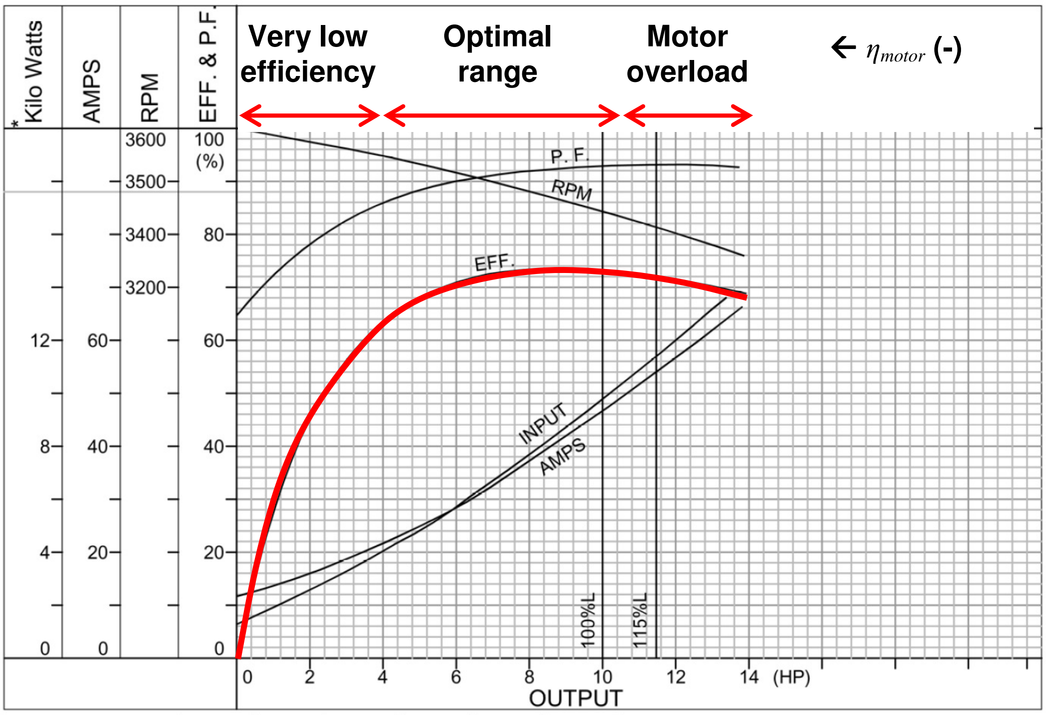
Some of the energy is also lost in order to cool the motor by the surrounding water. The motor efficiency is essentially independent of speed (BPMA 2002), but is highly dependent on the rated power (Table 2). Moreover, it is highly dependent on motor load, as shown on Fig. 13, especially for low loads (< 20% of power capacity) (U. S. Department of Energy 2007). Larger motors generally have flatter efficiency curves, and can be used at a satisfactory efficiency down to a 25%-load. Remarkably, the highest motor efficiency is often obtained for 75-95% load, where η_{motor} can be up to 0.5-2% higher than for 100% load (Bauer 2005). For the motor displayed on Fig. 13, the highest efficiency is obtained for a 85-90% load. This can be an advantage since motors are generally slightly over-dimensioned.

Table 2: rated motor power range and typical efficiency range, from (Mutschmann and Stimmelmayr 2007) and manufacturer communications.

Power (kW)	1	5	10	20	50	100
Motor efficiency η_{motor} (-)	63-75%	75-85%	80-87%	82-88%	86-90%	89-92%

2.2.6 Drive efficiency

The drive efficiency is related to the electronic components of the electronic drive; potentially including a frequency converter in case of variable-speed drives (Section 3.1.2). Depending on the position of the operating point on the system curve, it may in some circumstances lower the overall system efficiency by up to 5% (dena 2010). If the motor is used at fixed speed, then η_{drive} > 0.97 for most modern electrical drives (Hovstadius, Tutterow et al. 2005).



LOAD (HP)	NO LOAD (---)	25% (2.5)	50% (5)	75% (7.5)	100% (10)	115% (11.5)	125% (12.5)
AMPS	12.0	18.0	25.5	35.5	48.0	54.0	59.0
EFF.	0	54.0	67.8	73.1	73.6	72.0	71.5
P. F.	66.0	81.8	88.9	91.8	93.2	93.5	93.7
RPM	3600	3574	3538	3499	3452	3412	3398
* WATTS	1544	3454	5501	7654	10135	11830	13040

Full Load Torque 15.23 Foot Pounds
Break Down Torque 27.41 Foot Pounds
Locked Rotor Torque 21.32 Foot Pounds
Locked Rotor Current 202 Amperes
KVA Code E

Fig. 13: example of a submersible motor performance curve (Sterrett 2007).

Table 3 gives a summary of the different efficiencies of submersible pump system, the calculated global efficiency range and the typical values found in the literature.

Table 3: summary of efficiencies for submersible pump systems (derived from the figures given in Sections 2.2.4-2.2.6 and own calculations).

Efficiency	Depends upon	Range
Pump efficiency η_{pump} (-)	- rated power - impeller type and diameter - load	65-85%
Motor efficiency η_{motor} (-)	- manufacturing quality - cooling performance - load	75-90%
Drive efficiency η_{drive} (-)	- quality of electronic components - load	95-99%
Calculated range for η_{global} (-)	All above mentioned factors	46-76%
Literature values for η_{global} (-)*		45-73%

**(Mutschmann and Stimmelmayr 2007)*

2.3 Influence of pump and motor ageing on efficiency

Pumps, motors and pipes are sensitive to ageing. Among these, abrasive wear and corrosion are the most critical ageing patterns, both related to the quality of pumped water: abrasion is mostly caused by suspended sand particles. Sand is strongly abrasive; typically, industrial or municipal water systems can tolerate up to 5 mg/L sand without being impacted by abrasion (Borch and Smith 1993). Corrosion is especially frequent in water with high electrical conductivity or high chloride contents, and is enhanced if construction materials of the pump are alloys or low-quality metals and in conditions of long standby times.

The pump rotating speed (and thus discharge rate) and impeller design influence the resistance to abrasion. Ideally, materials having a fine grain structure shall be favoured for the pump components (Sterrett 2007).

Motors also age, and may fail for various reasons. One of the most limiting factors is the thermal insulation system and motor cooling. Overloading (and thus overheating) the motor is the most frequent reason for failure (Cornell Pump Company 2007).

In general, ageing induces more energy losses in the form of hydraulic friction, water recirculation, heat or mechanical losses.

All the above processes lower the efficiency of the pump and motor system, but will not be detailed further in this first phase of the project OptiWells.

Chapter 3
Options for an energy-efficient water abstraction

3.1 Adaptation of pump performance to reach higher efficiencies

3.1.1 Benefits of performance variation on efficiency

Pumps are often installed to cover a given range of discharges, and sized to satisfy the greatest demand. In this context, there is an opportunity to achieve energy consumption savings, especially during periods of low demand by adapting the pump's performance to the actual required performance (BPMA 2002). Although a slight oversizing of the equipment might be beneficial (motors are usually oversized by up to 15%, see Section 2.2.2), a pump system would need to be adapted if its operating flow rate varies by more than 10–20% of its BEP flow rate (Sustainability Victoria 2009).

The relevance of pump performance adaptation is related to the system's temporal distribution of flow demand, or system load profile. When the flow demand is relatively constant throughout the year, it is not relevant to use adapted pump performance methods. The load profile is represented by the distribution curve showing how often a given flowrate is exceeded. Fig. 14 shows typical examples of possible distributions. Depending on the number of operating points, the system is said to be *steady* (one operating point) or *variable* (several operating points).

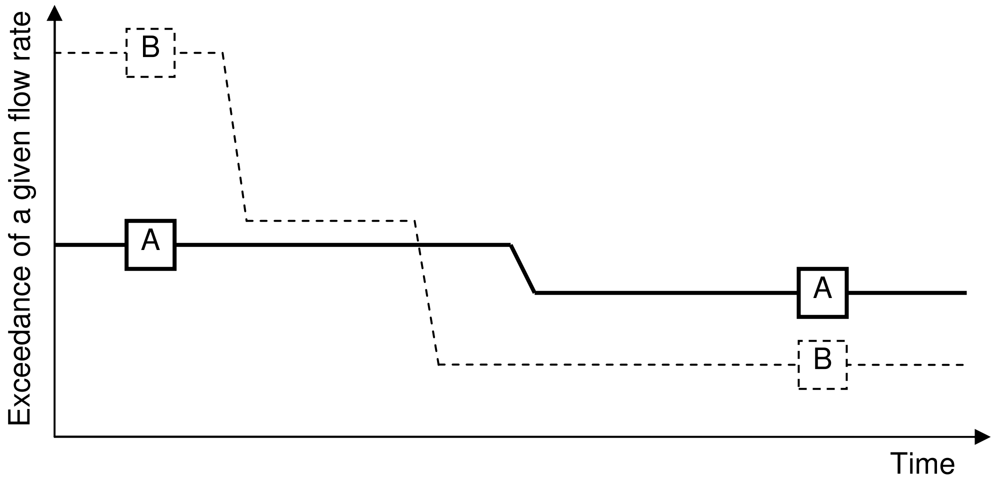


Fig. 14: examples of possible load profiles – adapted after (Hovstadius, Tutterow et al. 2005) (not to scale).

Load profile A shows only slight variations of demand, and no performance adaptation is required if the pump is dimensioned to operate efficiently at the higher flow requirement. This is typically a *nearly-steady system* with two close operating points.

Load profile B shows considerable variations in flow demand and adaptation of pump performance is necessary and definitely more efficient than using a pump sized for the peak demand at low load. This is a *variable system* with three significantly different operating points. A solution could be to use variable speed drives or several fixed-flow pumps in parallel to be able to meet the different demand levels.

The methods to allow performance variation include:

- pump control methods, allowing a high degree of flexibility (*variable speed drive*, flow-control valves / throttling, by-pass control, parallel operation of pumps, smart well field management);
- pump adjustments, which are less flexible or even permanent (*impeller trimming, adaptation of impeller stages*, pump exchange).

Throttling and by-pass control remain the most used means to adapt pump performance in most of the small- and middle-size installations (Gülich 2010). These types of performance adaptation are however inefficient, since the energy demand of the pump system is not lowered while its performance is lowered.

The methods for performance adjustment often rely on the affinity laws. These express ratios in discharge, total TDH and power as a function of rotation speed N or diameter D for the initial equipment (index "1") and the adjusted equipment (index "2") (BPMA 2002; Mosé and Roche 2005):

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2},\ \frac{H_{n,1}}{H_{n,2}} = \left(\frac{N_1}{N_2}\right)^2 \text{ and } \frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

In this section, focus will be given to methods that are relevant in terms of energy savings: *variable speed drive, impeller trimming* and *adaptation of impeller stages*. The combination of different pumps in parallel and the optimised operation of a well field, which are beyond the scope of this document, will not be considered here (this will be addressed by work packages 1 and 3 of the project OptiWells). The exchange of the pump is of course a method to adjust performance, but more cost-intensive and less flexible than the other methods this Section is addressing.

It must be noted however that, especially for large well fields, *parallel operation of pumps* and *smart well field management* are very relevant options for improving the efficiency at the scale of the well field.

→ *To find more information on the potentials of these techniques, please refer to the OptiWells-1-deliverable D 4.1 (Synthesis report).*

3.1.2 Variable speed drives

The basic principle of variable speed drives (VSD) is the ability to change the frequency provided to the motor (Sterrett 2007). VSDs are often used to correct for over-sizing of pump systems, and they have experienced a sharp increase in usage in the last decades (Hovstadius, Tutterow et al. 2005). However, their use for submersible pumps remains limited to date due to high static heads.

For induction motors, if a constant ratio of voltage to frequency can be maintained, then the motor's output rotating speed can be changed easily (Sterrett 2007). Indeed, according to Section 2.2.2:

$$\frac{f_1}{f_2} = \frac{N_1}{N_2}$$

In other words, when the frequency (and voltage) changes (keeping a constant frequency-voltage ratio), the motor changes its rotating speed in direct proportion to the change in frequency. This can lead to important energy savings, especially for low static head[7] and dissipative systems: the affinity law for centrifugal pumps shows that the power of the pump depends on the cube of the pump speed ($P \sim N^3$). At 50% flow, the necessary power is theoretically 0.5^3=12.5% of the power necessary at the full-flow operating point. Steep pump and system curves are particularly interesting for VSDs, since it is possible to achieve higher energy savings (Gülich 2010).

Benefits of VSDs include:

[7] i.e. systems where static head is significantly lower than the dynamic head at nominal discharge.

- a higher degree of flexibility in the operation of a well field. Depending on the system curve, they may help adjusting the operating point as close as possible to a new BEP, and save considerable amounts of energy;
- another benefit of VSDs is their ability to smoothly increase or decrease the speed of starts and stops. This enables to minimize inrush currents and hydraulic transients, thus protecting the system components (Sterrett 2007). This function may also be achieved by soft-starters.

VSDs also have some drawbacks:

- the electronic frequency converter has an own efficiency which needs to be considered in the efficiency of the drive, lowering its efficiency up to 5% (Hovstadius, Tutterow et al. 2005);
- for given pump curves with high static head, the use of VSDs may be counter-efficient since the change in frequency may result in a lower pump efficiency (shift of the pump curve) (BPMA 2002);
- the issue of motor cooling can be critical if the water flow is limited, and pumps may overheat and age faster – up to 30% less lifetime is reported (Sustainability Victoria 2009). What is more, a minimum speed must be maintained in order to keep sufficient lubrication of the bearings.

Variable-speed driven pumps shall not be considered as an ideal ready-to-use solution (as often praised by the manufacturers) without a careful examination of the system characteristics (pump curve, system curve, load profile…). In well fields, they are recommended to be used if flow rates vary over time by more than 30% (Sustainability Victoria 2009), and with a low geometrical elevation and a relatively long pipe network. However, when the speed of the pumps is modified, the TDH is also affected and it must be verified that VSDs can deliver a sufficient TDH for all speeds of operation. Besides, VFDs should not be operated under 50% of the nominal power frequency. The capital expenditure related to VSDs is relatively important.

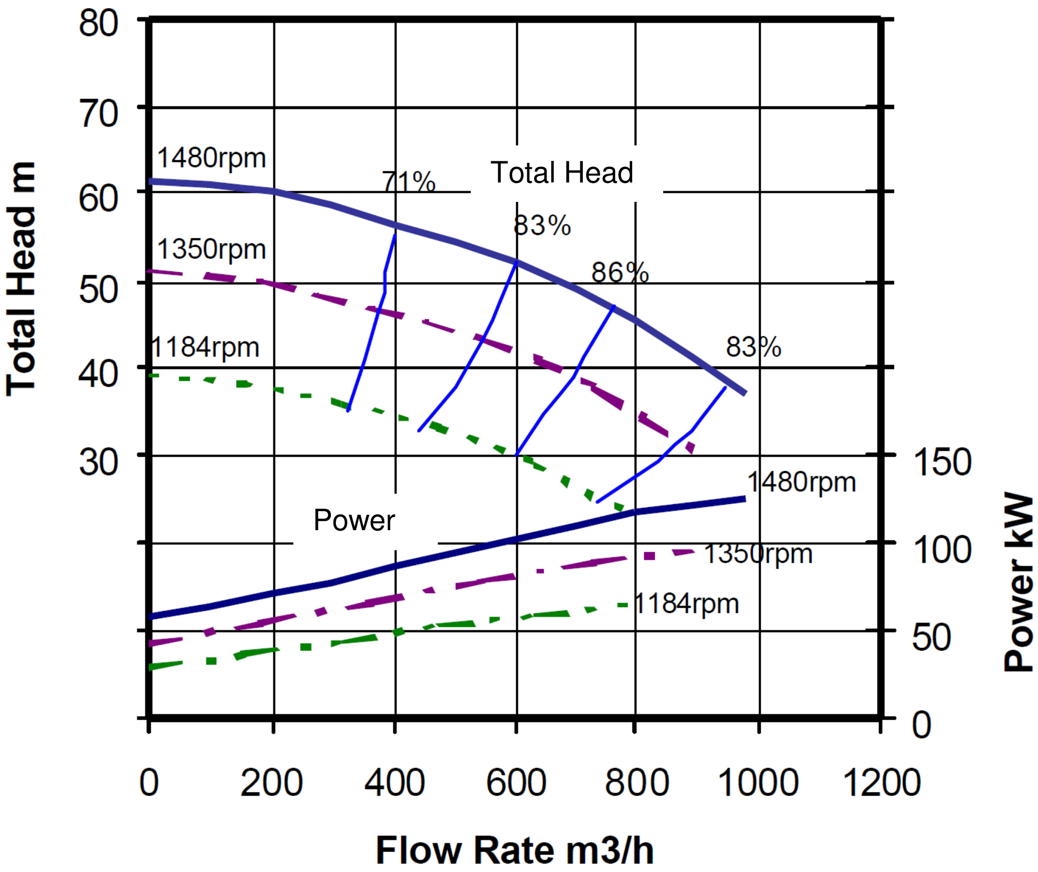


Fig. 15: effect of speed variation on pump performance (BPMA 2002).

3.1.3 Impeller trimming

The change in impeller diameter is a means to modify the pump curve. Impeller trimming is a common and cost-effective method for the adjustment of pump performance that requires machining of the impeller to reduce its diameter (Šavar, Kozmar et al. 2009). According to Section 3.1.1, the change in diameter shifts the pump curve in terms of discharge, and the following affinity is generally used[8]:

$$\left(\frac{D_1}{D_2}\right)^2 = \frac{Q_1}{Q_2}$$

The major benefit of impeller diameter change is that it may enable to lower the power demand of initially oversized pumps. However, the major drawback is that impeller trimming is not a reversible and flexible method of performance adjustment. It also requires the pump to be removed and dismantled.

Moreover, the reduction in impeller diameter shall not exceed 15-20% in order to avoid an overload of the impeller blades, turbulence and excessive noise (Gülich 2010). Excessive trimming can change the hydraulic conditions within the pump and lead to instability in operation and reduction in efficiency (EUROPUMP 2008).

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[8] although the affinity laws are generally used to determine the new pump curve, this is only an approximation, since diameter reduction leads to a new impeller that is not geometrically identical to the initial impeller (only the diameter is reduced, not the other geometrical characteristics). At the same time, similarity conditions are satisfied in many elements when impellers are trimmed, explaining why affinity laws are used for trimming limited to 15-20% (Šavar et al., 2009).

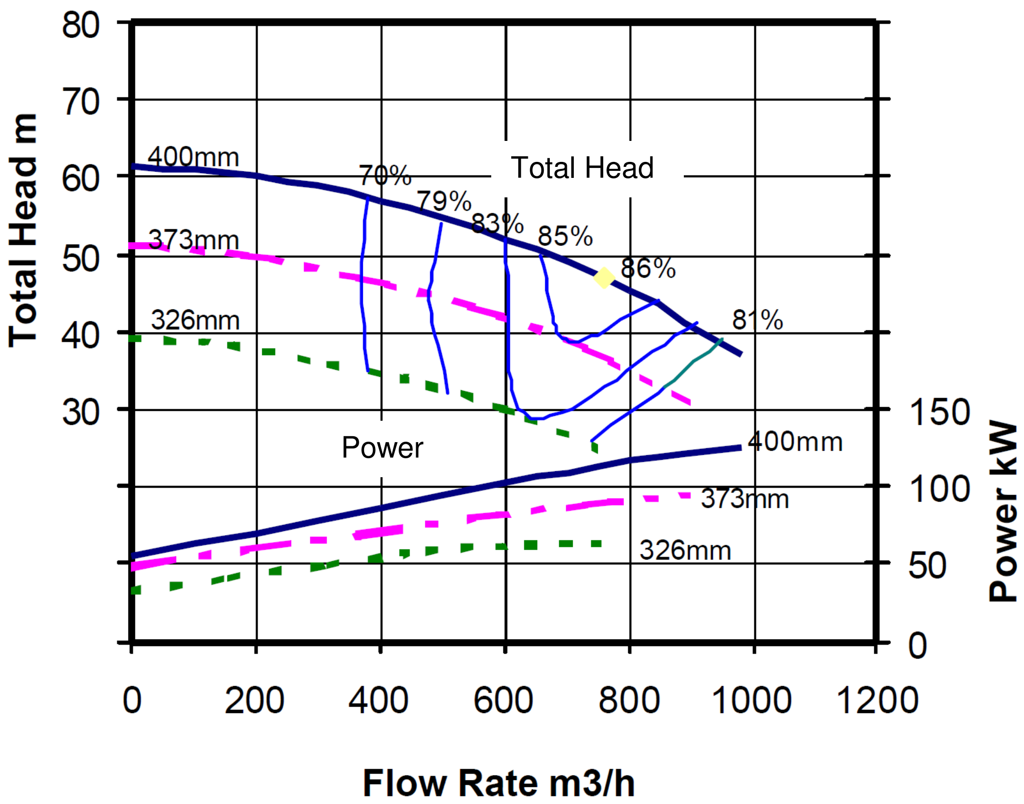

Fig. 16: effect of change in impeller diameter on pump performance (BPMA 2002).

Thus, impeller diameters shall only be altered to adjust to relatively small, permanent changes in the demand. Trimming remains a low-cost method of adjustment in comparison to the use of VSDs, but only if there are permanent changes in demand.

3.1.4 Adaptation of impeller stages

Most of the submersible pumps are multi-staged. This association of several stages works in a similar way to pumps in serial, and the overall pump TDH H for an n-stage submersible pump is given by:

$$H(Q) = \sum_{i=1}^{n} H_s(Q) = n \times H_s(Q)$$

with H_s the head for one stage. By varying the number of stages, it is possible to adapt the desired total TDH of the pump. Several pump manufacturers propose such types of modular stages which enable flexibility with short dismounting and reassembling, some even with discontinuous and modular shafts, to avoid dummy stages (e.g. (RITZ PUMPENFABRIK GmbH 2009)). However, this option is cost-intensive.

The drawback of this adaptation of the pump is that it requires dismounting of the pump, and usually the motor needs to be exchanged when changing the number of impeller stages, inducing additional costs.

3.2 Improvement of the pump system

3.2.1 Improvement in pump design

Most improvements that can be made in pump design are related to the choice of new, smoother components and / or coating the pump in order to minimize hydraulic resistance and head losses in the pump impeller and casing (dena 2010).

New materials, such as specific cast irons, bronze, stainless steel, show a much smoother surface than usual cast. Composite materials can also be used for that purpose. On the other hand, existing pumps can be smoothed to improve the pump efficiency, e.g. by coating the inner surfaces (impeller surface – see Fig. 17, pump shaft, or casing). The expected improvements in pump efficiency are given in Table 4 according to (Ludwig 2010). However, these represent potential maximum improvements, and may not be achievable for every pumping system.

Table 4: potential efficiency improvement by smoothing parts of the pump system – adapted from (Ludwig 2010). These are potential maximum efficiency improvements for a typical pump with n = 1450 min^{-1}, Q = 180 m^3/h.

Part to be smoothed	Impeller (inner side)	Impeller (outer side)	Casing	*Altogether*
Potential efficiency improvement $\Delta\eta_{pump}$ (-)	1.5%	6.5%	4.5%	*12.5%*

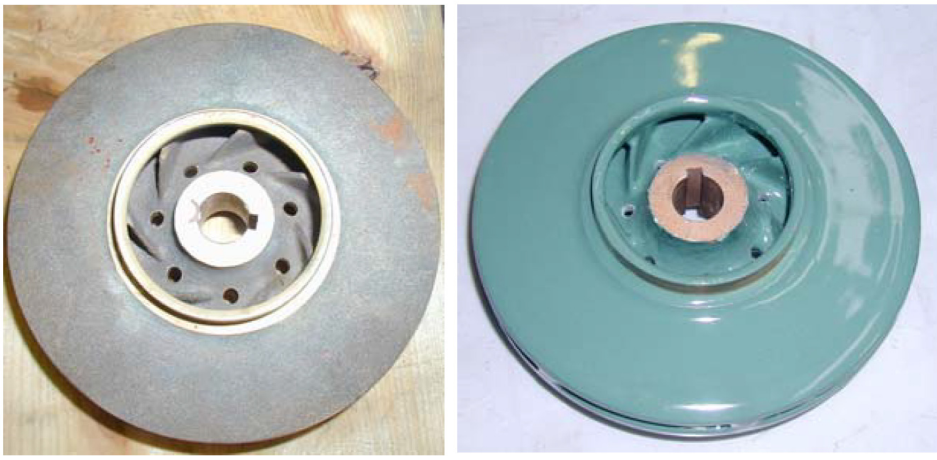

Fig. 17: uncoated and coated centrifugal, mixed flow impeller (Gudbjerg and Andersen 2005).

Pump coating not only improves energy efficiency, but it may also prolong the service life of the pump (Gudbjerg and Andersen 2005). Since drinking water requires high security standards, the applied coating material must be non-toxic and durable, meeting specific quality requirements (in France, “ACS – Attestation de Conformité Sanitaire”, in Germany the recommendation “KTW – Kunststoffe im Trinkwasser”).

3.2.2 High-efficiency submersible motors

Modern energy-efficient motors can save up to 10-20% compared to the motors currently in use in Europe (dena 2010). However, the stringent recent European norms for energy-efficient motors *do not apply to submersible pump motors*, which are embedded with the pump in a unique casing or body (Plath, Wichmann et al. 2010).

Thus, these high-efficiency classes are here presented only to show the possible future reference for submersible motors. Indeed, in the field of submersible motors, the manufacturers are developing high-efficiency motors that comply with the new energy efficiency classes IE-1, IE-2 and IE-3 as per EN 60034-30, although these do normally not apply to submersible motors. For a 30 kW-motor, this can bring significant improvements, and efficiencies of up to 93% can be reached (Table 5).

Table 5: former and new efficiency classes for squirrel-cage motors (BPMA 2002; dena 2010). Examples of motor efficiencies are given for a 30 kW aboveground asynchronous motor, f = 50 Hz, at full load.

New IE-Efficiency classes			**Former IE-Efficiency classes (CEMEP)**		
Efficiency class	**IE-Code and η_{motor}**		**Efficiency class**	**Name and η_{motor}**	
*Super Premium**	*IE-4**	*95.0%**	-	-	
Premium	IE-3	93.3%	-	-	
High	IE-2	92.5%	High	EFF-1	92.8%
Standard	IE-1	91.4%	Improved	EFF-2	92.0%
Below standard	-		Normal	EFF-3	below

**not yet available on the market at present date (even for aboveground motors)*

High-efficiency motors are designed to minimize inner motors losses such as (see also Section 2.2.5) (Kaya, Yagmur et al. 2008):

- losses through magnetic hysteresis and eddy currents in the stator cores. This can be done by increasing the number of laminations per kW drive rating, and by optimizing the stator windings for the respective rating and voltage;
- friction losses in bearing assemblies and mechanical seals. This can be done by using high-quality components for bearings, lubrication and sealing;
- heat losses in the stator winding and squirrel cage of the rotor. This can be done by lowering the rated current and the electric resistance to reduce losses, by using more copper per kW drive rating or increasing the cross-section of the copper conductors, and thus lower the average winding temperature;
- energy losses for coolant circulation. This can be done by reducing the required coolant quantity, or, if possible, to rely solely on the convection of surrounding water. This needs to be carefully evaluated, since sufficient cooling is essential.

These measures also have some positive side-effects (such as an improved Power Factor) and lower operating temperatures, which optimizes bearing lubrication and service life. Progress has been also made in assembling techniques, improving the general manufacturing quality of pump motors.

The combination of all these measures can typically improve the motor efficiency by 1-3%, although a further improve seems technically feasible in the coming decades, if innovations from the aboveground motors are transferred and adapted to submersible motors (BPMA 2002). The improvements in efficiency may be even higher for motors operating at lower loads, and for motors of low rated power (P < 20 kW), which are often used for shallow or small wells (Kaya, Yagmur et al. 2008).

3.2.3 Minimisation of hydraulic losses

Hydraulic losses do not directly influence the pump system efficiency, but result in wasted energy due to dissipation of energy in head losses, and change the operating point of the system. Apart from improvements in the pump design (Section 3.2.1), which can decrease hydraulic losses in the pump itself, it is also possible to adapt the hydraulic equipment, for instance:

- pump systems usually include a check valve to keep the pipe full of water when the pump is switched off. However, very often well heads also include check valves, resulting in an additional head loss which can reach up to 1 m water column. According to (Plath and Wichmann 2008), it is generally possible, without damage, to remove the well check valve and to reduce the hydraulic friction in the drop pipe accordingly;
- several other hydraulic equipments are located just downstream the pump, e.g. valves and pipe bends. These induce head losses which are proportional to the square of flow speed, which may be mitigated if appropriately designed:
    - for instance, pipe diameter D highly influences major losses, which are proportional to D^5 and pipes must be hence well dimensioned;
    - the replacement of traditional 90°-elbows by elbows with longer radius enables to divide the loss coefficient ζ by a factor 2 (Mutschmann and Stimmelmayr 2007);
    - the replacement of globe valves by gate valves (whenever possible) or of butterfly valves by new low pressure butterfly valves can divide the loss coefficient ζ by a factor 4, or even more.

Although not always directly related to the pump equipment itself, energy can be saved by examining the entire hydraulic system and adapting the equipment if relevant. This can be done, for instance, during a pipe maintenance.

3.3 Importance of pump and pipe maintenance

Pump and pipe maintenance is essential in order to maintain a high efficiency of the pump systems. According to (Mosé and Roche 2005), head losses in the pipes can be multiplied by a factor 4 to 10 due to ageing. In practice, head losses were for instance multiplied by 2 in a survey of a North-German water utility compared to the initial pipes (Plückers 2009), and also by a factor 2 at another German utility on 500mm-diameter pipes, 18 months after start of operation (Peters 2010). On the other hand, through pump maintenance, up to 10-12% can be gained in efficiency (Kaya, Yagmur et al. 2008).

The adequate maintenance nature and interval for the pump and the pipe system depends on the water quality, ageing patterns and well or pump operation, and is very site-specific. These aspects will not be detailed further in this first phase of the project OptiWells, but they need to be considered for an efficient well operation.

Chapter 4
Summary of energy saving potentials and conclusion

There is a significant potential for optimizing pump systems currently in use in groundwater wells. This potential lies in (Table 6):

- the improvement in *pump technology*, which can yield up to ~5% more efficiency;
- the improvement in *motor technology*, which can yield up to ~3% more efficiency, with further improvements if innovations from aboveground motors are adapted;
- the improvement in *performance adaptability*, which can be very efficient in some cases (~10-50%), but also counterproductive if not adapted to current situation (0% or even efficiency loss), and sometimes not very flexible (impeller trimming);
- the improvement of the *system maintenance* and *management* which may yield up to ~20% more efficiency, and which, in general, has a shorter payback time than performance adaptability options.

The improvement of equipments may induce only moderate additional costs if it is done at the time of scheduled new investments, after amortization of the equipment formerly in use. Unfortunately, these expected savings are influenced by uncertainties, which can be of the same order of magnitude as the savings themselves. For instance, the determination of the optimal operation point of a pump bears uncertainties between 1% and 4% and grows with pump rotation speed (Gülich 2010).

Other considerable saving potentials lie within cleaning, maintenance and smart well-field operation with short to moderate payback times (Table 6). These potentials are however very site-specific, and difficult to estimate on a general basis. Best practices for a “smart” pumping shall include choosing equipment that fits the actual requirements of the system, operating the pumps nearest of their Best Efficiency Point, and operating the motors in an energy-efficient load range.

The most obvious energy savings are those associated with improvements in the efficiency of the motor and of the pump (Shiels 1998). Such gains are often worth the added capital expenditure – although often having moderate to long payback times. However, as underlined by (Kaya, Yagmur et al. 2008), *that pumps have high efficiency alone is not enough for a pump system to work in maximum efficiency*.

An improvement of pump technology will yield, even optimistically seen, an efficiency improvement of up to 10%, which is the potential “theoretical limit” (EC 2003). For further improvements, it is necessary to consider solutions that go beyond the pump system, since maximizing efficiency depends not only on a good *pump* design, but also on a good *system* design. Even the most efficient pump in a system that has been wrongly designed is going to be inefficient. Moreover, an efficient pump in an inefficient well is pointless. Hence, a *global approach* of the groundwater abstraction system is required.

The optimization potentials *highly depend on the site characteristics themselves, on the local demand* (what distribution of the demand? what load profile?), and *on the operation and maintenance history* (e.g., what is the cleaning frequency of the pipes, if any?). Finally, one should not forget the primary objective of water abstraction, which is satisfying a given water demand, thus, the safety of drinking water production prevails over energy efficiency.

→ *To find more information on current innovations and levels of efficiency for available groundwater pumps, please refer to the OptiWells-1-deliverable D 2.2 (International market review of pumps available for groundwater wells).*

Table 6: summary of energy saving potentials (after (Shiels 1998; BPMA 2002; Schofield 2005; Kaya, Yagmur et al. 2008; Haakh 2009; Sustainability Victoria 2009; Boldt 2010), with estimates from current study).

Nature of improvement	Expected savings	Cost level*	Payback time**	Remarks
Improvement of pump and motor				
New pump technology	1 – 5%	€ - €€€	🕐🕐	If integrated in pump system renewal costs, the additional cost is moderate
New motor technology	1 – 3%	€ - €€€	🕐🕐🕐	
Performance adaptation				
Impeller trimming	0 – 20%	€	🕐	Only permanent downgrading
Adaptation of impeller stages / Modular shaft	0 – 10%	€€	🕐🕐	Few references, motor to be adapted
Variable speed drive to replace throttling	-10 – 50%	€€	🕐 - 🕐🕐	Very site-specific, not for high static head
System general improvement				
Pump cleaning	0 – 12%	€ - €€	🕐 - 🕐🕐	Also increases the pump lifetime
Pipe cleaning	0 – 10%	€ - €€	🕐 - 🕐🕐	Few references, depends on pipe age
Smart well-field management	10 – 20%	€	🕐	Few references, very site-specific

**€: low, €€: medium, €€€: high*

***🕐: short (2 years), 🕐🕐: medium (5 years), 🕐🕐🕐: long (10 years)*

Please note that VSDs can also cause increases in costs, explaining the potential negative savings. The values are given from the literature and/or from operators' personal communication. They represent average saving potentials for existing sites, but do not consider cases with obvious maladaptation of original design, where of course huge savings can be obtained.

Chapter 5
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Appendix
Calculation of major losses

The Darcy-Weisbach equation states that:

$$j_M = \lambda \times \frac{L}{D} \times \frac{V^2}{2 \times g}$$

where L is the pipe length, and D its inner diameter. To calculate the major losses, the unknown is the coefficient λ (Darcy friction factor), which is dependent upon the flow Reynolds number (Re), the roughness (k) and inner diameter (D) of the pipe.

For laminar flows (Re < 2000), the Darcy friction factor is only dependent on Re, and the Hagen-Poiseuille equation can be used:

$$\lambda = \frac{64}{Re}$$

For turbulent flows (Re > 4000, which is the case most of the time in pipe hydraulics), the Darcy friction factor depends on Re and the relative roughness (k/D). Several authors investigated this relationship for the different turbulent regimes, with the Colebrook and White formula being the most used:

$$\frac{1}{\sqrt{\lambda}} = -2 \times \log\left[\frac{2.51}{Re \times \sqrt{\lambda}} + \frac{k}{3.7 \times D}\right]$$

This implicit relationship is difficult to use analytically, and thus often the Moody-Stanton chart is used (Mosé and Roche 2005) (Fig. 18).

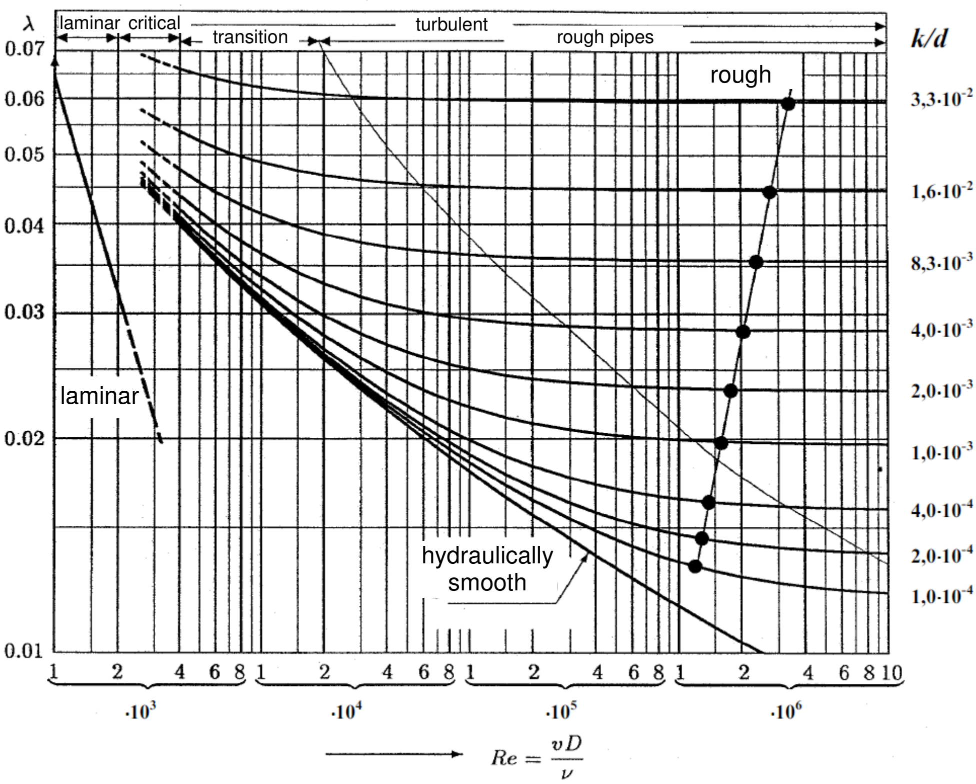

Fig. 18: Moody-Stanton chart to determine the Darcy friction factor λ (Strybny and Romberg 2007).