Energy-saving potential for centrifugal pump storage operation using optimized control schemes

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Abstract In this paper, we present the energy-saving potential of using optimized control for centrifugal pump-driven water storages. For this purpose, a Simulink pump-pipe-storage model is used. The equations and transfer function for steady-state and transient system behavior are presented and verified. Two different control strategies—optimum constant flow rate and level guided speed control—are compared to an allegedly optimal-driven pump with constant rotational speed. Twelve centrifugal pumps between 1 and 120 kW nominal power are evaluated to analyze the influence of different system parameters. The system characteristics, which are the static head, dynamic head factor, and maximum filling head, are varied 25 times for each pump in consideration of the pump's best efficiency point. Thus, 300 different systems are optimized for each control strategy and compared to the constant speed control. The results are analyzed and the relevant system's parameters, which have the most significant impact on energy savings, are shown. This

theoretical energy-saving potential is verified with measurements, which show the high impact of the part load losses of the frequency converter and the electric motor. A law for identifying and estimation potential energy savings is developed using this information. Four use cases are analyzed with this law. It is shown that for a cost-minimal operation, not only the savings potential but also the operating time is decisive.

Keywords Energy efficiency · Centrifugal pumps · Storage filling strategies · Pump system modeling · Control optimization

Introduction

Pump systems consume a large amount of the supplied electrical energy. According to estimates, the demand for energy is up to 20% of the total consumed electrical energy in industrialized nations (Kaya et al. 2008). If the focus is set on Germany, pumps cause about 12% of the electrical power consumption in the industrial sector, whereby annual savings of 33.6 PJ could be reached by increasing the efficiency (IFEU 2011).

Several capabilities can be applied to raise the total plant efficiency. The dynamic head loss can be reduced expensively by increasing diameter size. An additional measure is a retrofit of the installed pumps and drives using modern and efficient components, like electric motors with efficiency class IE3 or higher. By modifying the plant with a frequency converter, variable speed control can lead to energy savings between 30 and 50%

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after Kalaiselvan et al. (2016), the European Association of Pump Manufacturers (2004), Shankar et al. (2016), or Suh et al. (2015). This amount of savings can be reached by adapting the rotational speed and therefore reduce the flow resistance in the system. The savings depend strongly from the system structure, in particular the *pipe length*, *pipe diameter*, and the *static (geodetic) head*—what will be part of the discussion in this contribution. To control the speed of the motor frequency converters are increasingly used and are always getting more compact and cheaper after Lehmann (2009). Raising efficiency based purely on automation is a cost-effective way to decrease operating costs. It was also shown that the use of variable speed pump units not only results in lower pollutant emissions in operation compared to throttled pump units but also the manufacture of the respective required components leads to fewer negative environmental impacts in direct comparison to each other (Ferreira et al. 2011).

The actuators in these plants are mainly centrifugal pumps. As shown in European Commission (2001), 73% of the installed pumps are centrifugal pumps and the remaining smaller part belongs to displacement pumps. One of the most common applications for centrifugal pumps is to fill reservoirs. This use case takes place in water distribution systems or industrial plants like the petrochemistry or food industry.

In this kind of systems, Fig. 1 illustrates one exemplarily, a pump delivers a specific flow rate Q in $[m^3/h]$ to the storage. The pump generates a head H_P in $[m]$ that overcomes the static head H_S in $[m]$ and the dynamic head H_D in $[m]$ of the plant. Is the storage area A_s in $[m^2]$ of the filled storage tank finite and the filling process takes place from the bottom, like in Fig. 1, an additional filling level head H_L in $[m]$ increases during the filling time, as long as the discharge is small enough. Besides, a finite cross-section of the suction-side tank can mean a variation in the level there, which has the same effect on the pump.

Fig. 1 Basic topology of a pump storage system

Various target values can be defined for the optimization of storage systems. Thus, in addition to energy and cost-related optimizations, the running time and switching frequency can be taken into account. For this type of system, a primary distinction can be made between an optimal timetable of operation and an optimal controlled **operating mode**.

Electricity tariffs can be taken into account, and predictive timetables can be created to optimize the timetable after Prince and Ostfeld (2014). Also, predictive methods can be used to consider runtime optimization (Tang et al. 2014). Alternatively, the times of switching on and off can also be optimized in terms of cost-efficiency based on the filling level in Alvisi and Franchini (2016). These methods led to cost savings of between 12 and 30%.

An optimized control method, which is the subject of this contribution, can be achieved by adjusting the speed using a frequency converter. Santos and Seleglim (2005) and Hieninger and Schmidt-Vollus (2017) presented several speed control strategies, which lead to energy savings between 8 and 80% depending on the regarded system. The current filling level, based on the highest efficiency, adapts the rotational speed (Santos and Seleglim 2005). A similar strategy is shown in Hieninger and Schmidt-Vollus (2017), where this level-guided speed control is optimized during running operation and therefore considers the real plant behavior. This strategy optimizes the system while varying reservoir discharge is occurring. Another method is presented by Bene (2012), which finds control schedules for reservoirs using a mathematical model. This steady-state model was optimized with zero or constant discharge. Sometimes the filling time is an important parameter to fulfill proper and save plant operations. In combination with minimum energy demand, calculus of variations executes a flowrate control using in Lindstedt and Karvinen (2016). An alternative tuning method for a level guided speed control in combination with a maximum allowed filling time shows Hieninger et al. (2018). A method for more than one single pump for filling tasks developed Lücken et al. (2004), where a system of five pumps in parallel mode with a constant rotational speed per filling process was optimized. The authors reducing electrical energy costs with different tariffs and maintenance costs by Evolutionary Algorithms. Diniz et al. (2015) optimized a water reservoir driven by several pumps in parallel mode, which led to energy savings up 50%. The results of these

publications exhibit a broad distribution of energy-saving potential for running fluid storage in different ways. Different plant topologies cause this distribution.

The problems mentioned leads to the question, which parameters are decisive for energetically optimized operation of single pump systems. Typical parameters are the pump size, pipeline characteristics like the length, cross-section, friction and viscous losses, static head, and storage dimension.

Using the model of one pump, Ahonen et al. (2018) have shown the savings potential for optimum constant speed control– and level-guided speed control by varying the delivery head and frictional resistance of the pipe friction. Theoretical energy savings up to 50% in the part load area are possible by reducing the rotational speed. At higher a static heads, the level guided speed control leads to a higher saving potential than the optimal constant speed control.

The present contribution investigates the question of how the system properties affect the savings potential, whereby further parameters and optimized control variants are examined.

The interplay of the *static* and *dynamic head* at different *filling levels* and the influence of the *pump size* are analyzed. For these different parameters, two optimized control strategies, *level-guided speed control* (LC) and *flow rate control* (QC), are compared to the constant rotational speed operation with a simulation study. To realize this study, 12 different pumps, each with 25 different system characteristics, are simulated and optimized. The model of the pump system can calculate steady-state and transient pump behavior. The simulative results are only focused on the hydraulic systems because no information about varying efficiencies of frequency converters and the electrical drives are available in the datasheets of the regarded pumps.

Subsequently, the results are validated through a test and a *law for identification and estimation of theoretical energy savings* is presented. The law is estimating losses of the frequency converter and motor, which have an immense effect on the savings in the part load behavior. Using the law, energy savings for

- Single pump system
- Single pump system with storage filled from above
- Single pump system with storage filled from bottom or taken from the bottom
- QH or LC control
- Different pump size

can be determined. This law is applied to specific uses cases for estimating potential savings to demonstrate the application.

A plant operator can easily prove in pump audits using the law for energetically saving potential if energy savings can be identified. This method can also be used if a system is to be converted from a constantly operated to a speed-controlled pump unit.

The following section explains the methodology and how the systems are varied. Then, the equations and transfer function for the steady-state and transient description of the systems are derived and verified with experiments. Then, the different filling strategies are explained, which are then brought to their energetic optimum by an optimization algorithm. The results of potential energy savings, which are related to the operation at a constant speed, are shown based on the pump size and the system structure. Subsequently, a series of tests verify the results and, based on the conclusion, the law for estimating energy savings is developed. Finally, the law is applied to different applications in order to show the implementation by way of example and to demonstrate the necessary conditions for energy-efficient operation.

Methods

The energy efficiency of two different control strategies (see section *Filling Strategies*) is optimized for twelve different pumps in combination with 25 different plant characteristics. Thus, 600 different systems are optimized energetically. The results are compared to 300 systems using the standard pump operation at constant nominal rotational speed to show the energetic saving potential of optimized pump operation.

Pump variations

First, suitable centrifugal pumps have to be appropriately selected. This selection means that on the one hand, pumps of mid-range performance classes have to be covered in this study, which is the usually found performance class in industry and water supply, and on the other hand, pump characteristics have to be varied to detect distinctions in their energetic saving potential.

For this purpose, pumps with different tip speed ratios σ in [-] and diameter numbers δ in [-] are chosen (see Eqs. 1, 2). Small values of δ and bigger values of σ

characterizes axial pumps, analogously to that radial pumps. Diagonal machines are in the range between. The so-called *Cordier diagram* (see Fig. 2) illustrates the chosen tip speed ratios and diameter numbers, which is describing turbomachines in the *best efficiency point* (BEP) by the rotational speed n in $[\text{s}^{-1}]$, the flow rate Q_{BEP} in $[\text{m}^3\text{s}^{-1}]$, the delivery head H_{BEP} in $[\text{m}]$ and the impeller diameter D in $[\text{m}]$ in Cordier (1953). The specific rotational speed of the pumps lies between $7.7 \text{ min}^{-1} \leq n_q \leq 66.2 \text{ min}^{-1}$; thus, the shapes of the pump characteristics are similar but are compressed and stretched in their maximum values concerning volume flow, delivery head, and power (Gülich 2020).

$$\sigma = n \cdot \frac{\sqrt{Q_{\text{BEP}}}}{(2g \cdot H)^{\frac{3}{4}}} \cdot 2 \cdot \sqrt{\pi} \quad (1)$$

$$\delta = D \cdot \sqrt[4]{\frac{2g \cdot H_{\text{BEP}}}{Q_{\text{BEP}}^2}} \cdot \frac{\sqrt{\pi}}{2} \quad (2)$$

The *Etanorm* pump serves as a basis for the models. The nominal speeds are $n_n = 2900 \text{ min}^{-1}$. Furthermore, Table gives an overview. The performance of the pumps ranged between: $[1 \text{ kW} \leq P \leq 120 \text{ kW}]$.

System variations

For a sensible variation of the plant characteristics, all nominal operation points are set in the BEP of the respective pump. For each pump, five maximum filling heads H_L and five static heads H_S are chosen so that they match the pump characteristic curves (see Fig. 3).

The dimensionless dynamic head factor k_D , which contains information about the pipe length, pipe diameter, friction, and viscous losses, has to be adapted in that way, that the chosen H_L and H_S still fit the modeled

Fig. 2 Cordier diagram including chosen pumps

Fig. 3 Dynamic head factor and max. filling head variations in dependence of the static head

pumps. For that reason, k_D is calculated using Eq. 3 regarded to the BEP of each pump (see Fig. 3). For convenient dynamic losses, the flow velocity in pipelines should be between $1 \text{ m/s} \leq c \leq 2 \text{ m/s}$ (W400-1 n.d.). Therefore, the reference velocity is set to the lower speed limit of $c = 1 \text{ m/s}$.

$$k_D = \left(H_{\text{BEP}} - \frac{H_L}{2} - H_S \right) \cdot \frac{2g}{c^2} \quad (3)$$

Modeling

Each of the regarded pump systems consists of a pump, a pipeline, storage, and an added static head. Matlab/Simulink is used for modeling and Fig. 4 displays the basic structure of the model. The chosen control strategy gives the rotational speed n in $[\text{min}^{-1}]$ to the pump system. The pump model calculates the pump head. With this head, a one-dimensional flow in the pipeline is simulated, including steady-state and slow transient state behavior during a change in rotational speed. With calculating the pump power, the demanded energy per filling can be given. In the next steps, the model is explained in more detail.

Fig. 4 Basic signal flow diagram of the modeled pump driven storage systems

Affinity laws determine the pump head and the pump performance, whereby n corresponds to the nominal working point:

$$\frac{Q}{Q_n} = \frac{n}{n_n}, \quad (4)$$

$$\frac{H}{H_n} = \left(\frac{n}{n_n}\right)^2, \quad (5)$$

$$\frac{P}{P_n} = \left(\frac{n}{n_n}\right)^3. \quad (6)$$

Two polynomial functions with order j describe the specific pump characteristics with constant polynomial coefficients, which are calculated using the characteristics in the datasheet (KSB 2015). Calculating head H_P in [m] and power P_P in [W] is done using current flowrate Q , current rotational speed n , the polynomial coefficients A_i respective B_i and the affinity laws (Eqs. 4, 5, 6):

$$H_P = \sum_{i=0}^j \left(A_{i+1} \cdot Q^{j-i} \cdot \left(\frac{n}{n_n}\right)^{j-i-2} \right), \quad (7)$$

$$P_P = \sum_{i=0}^j \left(B_{i+1} \cdot Q^{j-i} \cdot \left(\frac{n}{n_n}\right)^{j-i-3} \right). \quad (8)$$

In the modeled system, these two equations represent two nonlinear gain functions without time behavior. The missing time behavior implies an immediate change in system response by change of input values and can be regarded as approximately accurate for energetic consideration, since the transient flow acceleration time t_T in [s] is much shorter than the filling time t_F in [s]: $t_T \ll t_F$. For more details about modeling transients within pumps, the authors recommend Dazin et al. (2015). The energy E_S in [Ws] to fill a storage is calculated employing the pump power (Eq. 8):

$$E_S = \int_{t=0}^{t=t_F} P_P \cdot dt. \quad (9)$$

To model the dynamic pipe and storage system, the energy conservation is used, where energy is equated with the delivery head. The current delivery head or pump head is equal to the total system head that is written by the *unsteady Bernoulli Equation* (Gülich 2020):

$$H_P = \frac{p_2 - p_1}{\rho g} + \frac{c_2^2 - c_1^2}{2g} + z_2 - z_1 + H_D + \frac{1}{g} \int_{l_1}^{l_2} \frac{dc}{dt} \cdot dl. \quad (10)$$

The first term on the right side considers a pressure difference in both reservoirs (see Fig. 1). In the case of modeling atmospheric storages, this term is set to zero. The second term regards differences in flow velocity c at the input and output of the system. Because the cross-section area of the storages is seen as relatively big, the difference in velocity is very low, and therefore, this term is set to zero. The third part of Eq. 10 represents the geodetic head, which describes the static head H_S and the filling head in the storage H_L . The flow rate and the cross-section of the storage A_R calculate the current level in the storage with:

$$H_L = \frac{1}{A_R} \int_{t=0}^{t=t_F} Q \cdot dt. \quad (11)$$

The dynamic head loss H_D regards friction and viscous losses in the pipe system. It is written as a function of the flow rate multiplied with the absolute value of the flow to consider negative flow rates:

$$H_D = k_D \cdot Q |Q|. \quad (12)$$

The dynamic head factor k_D is used to describe the nonlinear flow resistance. Theoretically, it implements the friction factor λ in [-], which depict the inner surface of the pipes, and the dimensionless loss coefficient ζ , which depict viscous effects of the fluid in combination with plant components and the pipeline structure. As a formula, the dynamic head factor is written with the *Darcy-Weisbach equation* (Brown 2003) as follows:

$$k_D = \lambda \cdot \frac{l \sqrt{\pi}}{2 \sqrt{A_P}} + \zeta. \quad (13)$$

The last term of Eq. 10 represents the unsteady part of the Bernoulli Equation, which is referred to the transient head H_T . This term describes the acceleration of the fluid with the velocity c within a pipe segment of the length of dl in [m], based on the linear momentum. Now the Bernoulli Equation can be formulated as follows:

$$H_P = H_L + H_S + H_D + H_T. \quad (14)$$

To have a look at the steady-state behavior of the model the transient head is set to $H_T = 0$ and the model is described with Eq. 15.

$$H_P = H_S + \frac{1}{A_R} \int_{t=0}^{t=t_F} Q \cdot dt + k_D \cdot Q|Q| \quad (15)$$

It is recognizable with Eq. 15 that for steady-state behavior, the filling process, the static head, and the dynamic head factor are decisive. The dynamic head factor is not implemented in the model using Eq. 13 to adjust these parameters to each simulated pump, k_D is adapted to the pumps of Table 1 with Eq. 3, so the pipe length and the pipe diameter are considered with one value regarded to the BEP of each pump.

In the next step, we built a transfer function that has to implement the transients in the model. First, the input head to the unsteady systems is written with $H_{PS} = H_P - H_S$. This is because the static head is a not time-dependent parameter. Otherwise, the transfer function will omit the static head. With Eq. 15 and the last term from Eq. 10 to describe the transients we got:

$$H_{PS} = k_D \cdot Q|Q| + \frac{1}{A_R} \int_{t=0}^{t=t_F} Q \cdot dt + \frac{1}{g} \frac{1}{l} \frac{dc}{dt} \cdot dl \quad (16)$$

The flow velocity is $c = Q/A_P$ and there is only one pipe segment by using a constant pipe cross-section area A_P , which leads to Eq. 17.

$$H_{PS} = k_D \cdot Q|Q| + \frac{1}{A_R} \int_{t=0}^{t=t_F} Q \cdot dt + \frac{l}{A_P \cdot g} \cdot \frac{dQ}{dt} \quad (17)$$

In the next step, we write system inputs on the right side and the output on the left side, $k_S = A_P \cdot g/l$ in $[s^2]$ and $\dot{Q} = dQ/dt$:

Table 1 BEP characteristics of chosen pumps (KSB 2015)

Pump no.	δ in [-]	σ in [-]	Q in $[m^3/h]$	H_P in $[m]$
1	16.3	0.064	4.7	22.5
2	21.23	0.049	6.8	41.4
3	6.85	0.156	16.3	15.8
4	15.27	0.063	24	69
5	17.99	0.057	16.2	60.5
6	17.26	0.057	39.5	109
7	3.25	0.342	80	16
8	4.49	0.223	72	26.5
9	6.72	0.145	70	46
10	11.28	0.087	96	113
11	2.76	0.419	285	28.5
12	5.83	0.171	342	106

$$k_D \cdot Q|Q| + \frac{1}{A_R} \int_{t=0}^{t=t_F} Q \cdot dt + \frac{1}{k_S} \cdot \dot{Q} = H_{PS} \quad (18)$$

In a normalized transfer function, no integrals are available, so Eq. 18 must be derived:

$$\ddot{Q} + 2k_D k_S \dot{Q}|Q| + \frac{k_S}{A_R} Q = k_S \cdot \dot{H}_{PS} \quad (19)$$

This transfer function for slow transient and steady-state flow is used for modeling in Matlab/Simulink, which structure shows in Fig. 6.

For a short model verification, Fig. 5 shows two step responses of the pump head, the flow rate, and the shaft power due to change in rotational speed related to the final value at $n = 2000 \text{ min}^{-1}$. As a reference, step responses are measured with a centrifugal pump (KSB 050-032-125-124). The input values for the model uses the measured rotational speed. The time behavior of the modeled flow rate and pump head fits very well with the measured ones. The shaft power shows a difference in time behavior (Fig. 6). The simplification of the model causes this difference by omitting a motor model. Thus, the power can only be calculated with usage of the flow rate. A motor model calculates the speed as a function of the load torque and the motor torque with a specific time behavior, which would agree with the measurements (cf. shaft power and speed in Fig. 5).

A steady-state model verification shows Fig. 7, which compares modeled and measured values of the pump head and shaft power of the pump to each other. As reference rotational speed the measured values for $n = 2012 \text{ min}^{-1}$ are taken and polynomial functions (Eqs. 7, 8) are built. It can be seen that the real pump produces less head at lower speeds with higher power consumption. The reason for this is that internal losses and mechanical friction increase in percentage terms at lower speeds (Gülich 2020).

Filling strategies

In general, three different control strategies can fill a reservoir:

- n_c : constant rotational speed control (NC)
- Q_c : constant flow rate control (QC)
- n_L : head guided speed control (LC)

Fig. 8 illustrates these strategies, the dynamic and static plant characteristics, and the movement of operation points between two filling levels. Here, the

Fig. 5 Step response for change of rotational speed from $n = 800 \text{ min}^{-1}$ to 1400 min^{-1} , 2000 min^{-1}

reservoir is filled from the bottom, so the effect of a varying filling level respective filling head to the hydraulic system is included. The start-up begins with a flow rate of $Q = 0 \%$. The dotted lines depict the transient head H_T during start-up period for the three control strategies, whereby a little overshoot for the QC strategy occurs. After reaching steady-state, what Fig. 8 shows exemplarily for the NC, the filling process starts. A little volume of fluid is already pumped into the storage during acceleration. Due to the short transient time compared to the long filling time, this amount of fluid in the storage at the beginning of the steady-state can be neglected and $H_L = 0 \text{ m}$. Furthermore, after reaching steady-state a small amount of transient head is still present, because of the varying storage level respective head $\dot{H}_L$.

Starting from this point, NC is filling the storage with a constant rotational speed along the QH-characteristic curve. The advantage of this method is that no additional sensors are required.

QC is operating at a constant flow rate, which is the constant reference flow rate in Fig. 8. The QC has the advantage that the process can be better controlled and monitored by recording and regulating the volume flow.

Fig. 6 Block diagram of the modeled pump system

Even with a high delivery head, QC guarantees safe operation.

NC or QC gives out directly the setpoints to run the system. By contrast, LC gives the setpoint indirectly by the current filling level added up with the static head. This control strategy has the advantage that it directly responds to the current storage state and thus adapts the speed to a variable optimum. Fig. 9 shows an exemplary process of speed due to an increasing filling level in consideration of a limited rotational speed $[n_{\min} \leq n_L \leq n_{\max}]$. The higher the filling level, the higher the speed.

Fig. 7 Modeled and measured pump characteristics for KSB 050-032-125-124

Fig. 8 Filling strategies (Hieninger and Schmidt-Vollus 2017)

To create a proper function for guiding the speed n_L in $[\text{min}^{-1}]$ in dependence of the head, the affinity laws are used:

$$n_L = n_n \cdot \sqrt{\frac{H_L + H_S + H_D}{H_{\text{BEP}}}}. \quad (20)$$

Here, H_{BEP} is the optimal head at the reference speed n_n . We use this equation to develop a control law for minimizing the dynamic head loss. Because the dynamic head loss is not known, we set it to zero in the control law: $H_D = 0$. Thus, the speed can be calculated with an offset to overcome the static head and with a varying speed set point caused by the varying storage level. This calculation method can be seen as a disadvantage compared to the QC, because the speed must always be high enough to handle the filling tasks. To handle this, an increased effort in the implementation is noticeable.

To tune the control strategies QC and LC by minimizing the target value w (Eq. 21) the control parameters v_i have to be adapted. Eq. 22 reflects the optimal set flow rate as a one-dimensional optimization problem. Eq. 23 is the control law for the LC, where a two-dimensional optimization problem is present. To calculate the optima for Eqs. 22, 23 by minimizing Eq. 21, the Matlab function *fminsearch* is used, which is also known as the *Nelder-Mead* algorithm (Lagarias et al. 1998).

$$w = \min(E) \quad (21)$$

$$Q_c = v_i \quad (22)$$

Fig. 9 LC with limitations

$$n_L = v_{i,1} \cdot (H_F + H_S)^{v_{i,2}} \quad (23)$$

Energy-saving Potential

This section shows and analyzes the results of the tuning process. First, the quality of the optimization algorithm is checked. Second, the potential energy savings in dependence of system parameters are presented and discussed. In the third part, the increased filling time, caused by the energy-efficient operation, is considered. In the fourth part, the results are verified and the influence of additional losses is analyzed.

Verification of optimization results

We choose two exemplary systems to verify the resulting optimized operation points for LC and QC. The reference values are determined using brute-force (BF), which tested the reference system with 50 different values for each $v_{i,1}$ and $v_{i,2}$, so in total 2500 reference values.

In Fig. 10, the resulting filling energy for LC using different tuned $v_{i,1}$ and $v_{i,2}$ is shown. Here, two optima occur, because two different starting points are preset to the *fminsearch* function in Matlab, which optimizes the model in Simulink. The locations of the detected optima are in the immediate surrounding of the BF optimum. Thus, the optimization can be seen as all right. In the white area, no filling is completed due to improper control parameters.

The BF method calculates the trend of the filling time respective the filling energy to check the QC. Here, the ascertained best operation points almost coincide (see Fig. 11).

Potential energy savings for LC or QC

Fig. 10 Filling energy using LC strategy at different control parameters $v_{i,1}$ and $v_{i,2}$

Fig. 12 illustrates the theoretical energy savings e in [%] using LC and QC. The optimized filling energy $E_{S,opt}$ for the LC and QC referred to the filling energy $E_{S,n}$ with constant rotational speed of $n = 2900 \text{ min}^{-1}$ defines the saving potential.

$$e = \left(1 - \frac{E_{S,opt}}{E_{S,n}} \right) \cdot 100\% \quad (24)$$

The results are presented in dependence of κ in $[\text{m}^{-1}]$, which describes the ratio between the dynamic head factor k_D , and the mean total static head during filling (see Eq. 25).

Fig. 11 Filling energy and time using QC strategy at different control parameters v_i

$$\kappa = \frac{k_D}{H_S + 0.5 \cdot H_L} \quad (25)$$

The red-dashed and dash-dotted lines represent functions of the reachable mean energy savings Δe . These functions can be reproduced by using the polynomial coefficients in Table 2. A differentiation between pumps with theoretical lower efficiencies and theoretical higher efficiencies is applied according to Eck (2003) to achieve more accurate results:

$$\sigma = \begin{cases} < 0.1, & \eta < 60 \\ \geq 0.1, & \eta \geq 60 \end{cases} \quad (26)$$

For pumps with higher σ and, therefore, with a theoretical higher η in [%], slightly larger savings Δe are possible. If no static head exists—solely horizontal piping—then $\kappa \rightarrow \infty$. In this case, the saving trend will hypothetically reach 100%.

Both control strategies lead nearly to the same energy savings, but LC is a little bit more efficient (see Δe at Fig. 13). Bigger values of H_L in these models leads to increase of savings.

For such, use cases can be concluded that the most important effect to save energy is to reduce the flow rate to a minimum until the partial load losses become correspondingly high. Lowering the speed at the beginning and raising the speed during the filling process causes this effect.

Due to the reduction of the flow rate, the pump unit is operating in the partial load. It is well known that the efficiency is decreasing in this area. This effect is exemplarily shown for the KSB 050-032-125-124 pump in Fig. 14, where the upper figure displays the efficiencies of the motor η_M , of the frequency converter η_{FC} and the combination of both η_{M+FC} . Likewise, the efficiencies which result from the hydraulic power in relation to the shaft power η_2 , to the electric motor power η_1 and to the electrical power supply including frequency converter losses η_0 are decreasing. However, an optimal filling process can be described not so much in terms of efficiency as in terms of a minimum of specific energy requirements E_{spec} in $[\text{Jm}^{-3}]$:

$$E_{spec} = P/Q \quad (27)$$

Concerning the filling process over the filling time, it results in filling energy related to the storage volume V_S in $[\text{m}^3]$, according to Eq. 28.

Fig. 12 Theoretical energy saving potential for LC and QC control (legend is split and valid for both)

$$\int_{t=0}^{t=t_F} E_{\text{spec}}(t) dt = E_S / V_S \quad (28)$$

In this way, a minimum of specific energy requirement integrated over time describes an optimum filling process. For just this parameter, Fig. 14 shows that there is a maximum (optimum) of $1/E_{\text{spec}}$ with regard to the delivery of volume flow, especially in the partial load operation area. Whereby it is shown, by consideration of additional losses, the optimum is shifted towards higher flow rates.

The decreased Q raises the filling times at higher filling efficiencies. Fig. 15 shows the dependence between saved energy and additional time. It can be seen, that mostly every energy-efficient filling leads to a raise in time. This is caused by reducing dynamic losses in the pipeline due to of smaller adapted flow rates.

Table 2 Polynomial coefficients for saving curves

	σ	k_1	k_2	k_3	k_4
LC	$\sigma \geq 0.1$	0.0001	-0.0255	2.1853	-0.5452
LC	$\sigma < 0.1$	0.0001	-0.0179	1.8367	-0.9558
QC	$\sigma \geq 0.1$	0.0001	-0.0243	2.1392	-0.6283
QC	$\sigma < 0.1$	0.0001	-0.0175	1.8188	-1.0746

Verification of the method for identifying energy-saving potential

This section should verify the proposed method to identify energy-saving potential in pump systems. For this purpose, system characteristics are measured with the KSB 050-032-125-124 pump ($\sigma = 0.145$ and $\eta = 66.2\%$), one with open valve and one with throttled valve, resulting in different dynamic head factors of $k_{D, \text{open}} \approx 34$ and $k_{D, \text{throttled}} \approx 71$ at a flow velocity of $c = 1 \text{ ms}^{-1}$, which are shown in the upper figure of

Fig. 13 Energy saving comparison of LC to QC in dependence of the filling level

Fig. 14 Part load efficiency and specific energy requirements

Fig. 16. The factor increases at a lower speed, so a nonlinear behavior of k_D can be detected. With an existing static head of about $H_S \approx 1$ m in the test facility and no difference in filling level $H_L = 0$ m energy savings can be calculated using Eq. 25 and the polynomial coefficients for QC from Table 2.

Regarding to the shaft power P_2 , the estimated savings are calculated with higher values than occurring in reality (Fig. 16, Table 3). The reason for this is the deviation of the affinity laws in the partial load range, which determines lower powers than occur (cf. Fig. 7). Further losses, which are taken into account in the motor efficiency and frequency converter efficiency, additionally reduce the real savings potential. Thus, the savings in this application case reach about one-third to one-quarter of the estimated values.

For this reason, we propose a reduction of the savings. If the plant has a relatively high dynamic component ($\kappa >$

Fig. 15 Additional filling time t_F^+ in dependence of the saved energy

20), it can be assumed that the plant is operated in an energetically optimal way in the partial load range, as has just been shown. In this case, a conservative reduction of $\kappa_{\text{Red}} = 1/4$ is proposed. If the plant has a high static component, the optimum will be closer to full load. Here, a reduction of $\kappa_{\text{Red}} = 1/2$ should take account of the losses. Thus, an estimation of the savings is made as follows:

Fig. 16 Verification measurement with two valve positions and for shaft power respective electrical power supply

Table 3 Comparison of estimated and measured savings

e in [%]	Estimated	Measured (P_2)	Measured (P_1)	Measured (P_0)
Valve: throttled	64.5	53.9	32.2	22.9
Valve: opened	47.9	38.9	18.7	12.1

$$e = \kappa_{\text{Red}} \cdot (k_1 \kappa^3 + k_2 \kappa^2 + k_3 \kappa + k_4). \quad (29)$$

Use cases

Four use cases are defined to show how this method can be used to determine energy-saving potential and the cost benefits of an optimized control strategy, which are shown in Table 4. The first and second use case describes a typical filling task, which may occur in the industry. There is little head to overcome, whereby the second application case is larger in its design. The third case describes a water tower for the water supply of a city. Here, the pump is located at the bottom of the tower. The static head that needs to be overcome takes up almost the entire length of the pipeline. The fourth case refers to a high-level storage facility, which is supplied via a long feeder line. Here, the change of the filling level is relatively small compared to the static head. All calculations are made approximately with zero discharge from the reservoirs.

The used pumps are shown in Table 1. The dynamic head factor is calculated with a pipe friction factor of $\lambda = 0.3$ mm and with a loss coefficient of $\zeta = 50$. Each pipe diameter is calculated so that the flow velocity is $c \approx 1 \text{ ms}^{-1}$ at Q_{BEP} . The fillings per day are estimated, whereby in industrial plants, more fillings are assumed than in the storages for water distribution. The additional costs for frequency converters depend on the pump power. The costs for engineering are seen as constant because the effort of PLC implementation and wiring are not strongly dependent on the pump size. Both types of costs are roughly assessed. All savings are calculated using Eq. 29. The cost of the electrical energy is set to 0.16 €/kWh, which is the tariff for industrial customers in Germany in April 2019 (Statista 2020).

The first use case has a small pump power. Thus, the cost savings are relatively low. Even with 15.2% energy savings, there is a negative cost balance after 10 years. The second use case represents the same plant scenario, and it is only larger in its dimension. Here, despite less

estimated energy savings, the larger design leads to a higher savings potential in operating costs. If, in this application scenario, the fillings per day are reduced to one filling, the additional costs are only amortized after 16 years. The amortization time shows that the operating hours of the pump must be taken into account to determine cost savings, even if there is a relatively high available energy-saving potential.

In the third use case, the storage is filled every fourth day, and because of the small κ small theoretical energy savings are calculated. Due to that, only after 20 years, the cost balance becomes positive. By gaining the fillings up to one filling every second day, after 10 years, the cost balance is positive.

The fourth use case has a medium theoretical energy-saving potential. The high-level reservoir is filled every second day, which can be seen as a typical value for this kind of tanks. After six and a half years, a positive cost balance can be seen.

Conclusions

In this contribution, a law to estimate energy savings for single pump systems for filling tasks is developed. These systems can have a varying head due to the filling process, but also savings for systems without this varying head can be calculated. 300 pump systems are modeled, optimized with two control strategies, and compared to the standard control running at nominal speed through the BEP to develop the estimation method. Both control strategies lead to nearly the same savings of up to 70%. It was found that the main positive impact on the savings is a high dynamic head loss compared to the static head, whereby a higher difference in filling levels leads to higher savings using LC method. Furthermore, a raised tip speed ratio and pump efficiency increase the potential energy saving for both control strategies. This method works independently of the pump size, rotational speed, and also of its design (axial, diagonal, radial) because the reference point of the expected savings is set to $c = 1 \text{ ms}^{-1}$.

The simulation study does not include increasing part load losses of frequency converter and motor. Additionally, affinity laws calculating less pump power than expected. For this reason, the estimation law is reduced in dependence on the static head.

The use cases show depending on κ significant theoretical energy savings and CO₂ reduction in single

Table 4 Cost balances for use cases

Examples:	Industrial storage		Water tower	High-level tank

	Pump factors			
Pump No.	3	11	6	6
Pump power P_{BEP} in [kW]	1.17	36.89	19.55	19.55
	Plant factors			
Static head H_s in [m]	1	1	50	100
Filling level H_L in [m]	3.5	5	25	3
Storage Cross Section A_R in [m ²]	25	75	50	100
Pipe length l in [m]	30	100	60	1000
Dynamic head factor k_D in [-]	170	143.75	200	2550
κ in [-]; κ_{Red}	61.8; 1/4	41.1; 1/4	3.2; 1/2	25.1; 1/2
Fillings per day	3	3	0.25	0.5
Filling time t_L in [h]	5.37	1.32	31.65	7.59
	Additional costs			
Costs: frequency converter [€]	300	3000	2500	2500
Costs: engineering [€]	3000	3000	3000	3000
	Life cycle costs			
Energy requirement per year in [kWh/a]	6875	53150	56466	27097
Energy costs per year [€/a]	1100	8504	9035	4336
Theoretical energy savings e in [%]	15.2	13.3	3.1	19.9
Theoretical cost savings per year in [€/a]	167	1129	280	864
Amortization time [a]	19.8	5.3	19.7	6.4
Cost balance after 10 years [€]	-1630	5294	-2704	3138
Cost balance after 20 years [€]	40	16587	92.3	11776

pump systems using the reduced estimation law. The major positive effects on cost savings are a high value of dynamic head related to the static head and the operation time of the pump.

Smart control strategies have to be applied in storage systems to reach this saving potential. These control strategies have to be developed to adapt control parameters to the real system behavior and guarantee safe fillings. A safe filling can be disturbed by high fluid demand combined with adapted control parameters, which lead to a long filling time.

Plant operators can use the results in Fig. 12 to estimate the potential energy savings. Due to deviations of the part load losses, the potential energy savings should be halved to give a conservative estimate. This decision support can help the operators to implement better control strategies. Additionally, they should look at the theoretical increase of filling time in Fig. 15 and compare the time to their system's maximum fluid demand. The control strategy must be adapted if there is a risk that the storage cannot be filled.

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Declarations

Conflict of interest The authors declare that they have no conflict of interest.

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