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The best attainable EEI for Booster Stations derived by Global Optimization

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Summary The Energy Efficiency Index (EEI) offers the possibility, to rate the efficiency of a booster station by taking a load profile into account. Besides the quality of the used single machines, the EEI is massively depending of the applied operational strategy. For most operational points different numbers of pumps are able to cover the actual load, but for only one of these scenarios the energy consumption becomes minimal. This paper presents an approach for calculating the maximal attainable EEI for booster stations from the characteristics of the machines. Only the data of the single pump unit within the booster station is needed to calculate the EEI of an optimally operated booster station. The result of the optimization program is the optimal operational setting for every point of the load profile and thus the best achievable EEI. Furthermore we use this approach to make proposals for possible improvements for the control and topology of booster stations.

1. Introduction

Pumps and pumping systems consume about 10 % of the annually produced electricity in Europe [1]. The European Union aims for the usage of energy efficient machines to gain energy savings. These energy efficient machines are the first step towards reducing the energy consumption of pumping systems. Therefore, the market of pumps and booster stations is going to be strictly regulated and energy labels will be introduced. Various eco-design directives will be carried out in order to ban machines with low efficiency from the market [2]. The preliminary Energy Efficiency Index (EEI) considers more than one duty point for the efficiency rating. The power input in the field of operation is rated by the EEI.

Booster stations meet a varying pressure demand with high energy-efficiency by deactivating individual pumps at smaller loads. They cover a wide performance range and are flexible machines with widespread possibilities of usage in industrial and residential applications, e.g. chemical processing plants or water supply in skyscrapers. The difference between single pump units and booster stations is the diversity of control options. Both, rotational speed and the number of active pumps, determine the operational point of the booster station and thus its energy efficiency. For most operational points, different numbers of pumps are able to cover the actual load, but for only one of these scenarios the power input becomes minimal. The operational strategy of the booster station may vary and massively influences the energy input and thus the EEI.

This paper presents an approach for calculating the best attainable EEI for booster stations based on the characteristics of the machines. We describe the technical problem and give the technical data for the example of the paper. Afterwards we formulate the technical restrictions, degrees of freedom and objective as an optimization program. The optimization program consists of the technical data of the machines, the physical restrictions and the load profile. Only the data of the single pump unit within the booster station is needed to calculate the EEI of an optimally controlled booster station, i.e. no control assumptions are needed. The result of the optimization program is the optimal control setting for every point of the EEI load profile and thus the best achievable EEI. Furthermore we use this approach to make suggestions for possible improvements of booster stations.

2. Technical Specifications

Working Principle of a Booster Station

The technical task of a booster station is to promote water in a piping network. Therefore the pressure is increased to overcome geodesic differences or pressure loss by dissipation. The connection scheme of a booster station is simple: Multiple pumps, in most applications of the same type, are connected in parallel. The incoming water flows from the suction pipe, through the single pump units, into the pressure pipe. The pressure at the outlet is always higher than at the inlet. The load of the booster station is given by the total volume flow Q_l and the pressure head ΔH_l. To avoid reverse flow, a check valve is installed behind each pump.

The parallel arrangement has two major reasons: Firstly, one is able to avoid heavy part load in the whole system by deactivating single pumps. Secondly, if one pump fails, it is still possible to use the others for promoting water. Other topologies, especially the direct connection of pumps in series, are generally possible and useful [3], but the application of an operational strategy in between the considered duty points is very difficult. Thus, we do not consider this topological option in our paper.

Pump Characteristics

To describe a pump's field of operation, four parameters are relevant: The volume flow Q, the pressure head ΔH, the rotational speed n and the power input P. For strictly monotonic characteristics the actual duty point is determined exactly by any two of these parameters. This condition is fulfilled for the relevant operating range for any pump and in this paper in the whole operating range. Figure *1* shows the characteristics of a single pump.

The manufacturer describes the characteristics in his catalogue based on measurement data. For the reference rotational speed n_{ref}, pressure head and volume flow of the pump are varied in a test rig and these values as well as the corresponding power input are measured. Following industrial standards the unit to measure the pressure head is meter water column (mWC). Table *1* gives the measurement points for the pump type A considered in this paper.

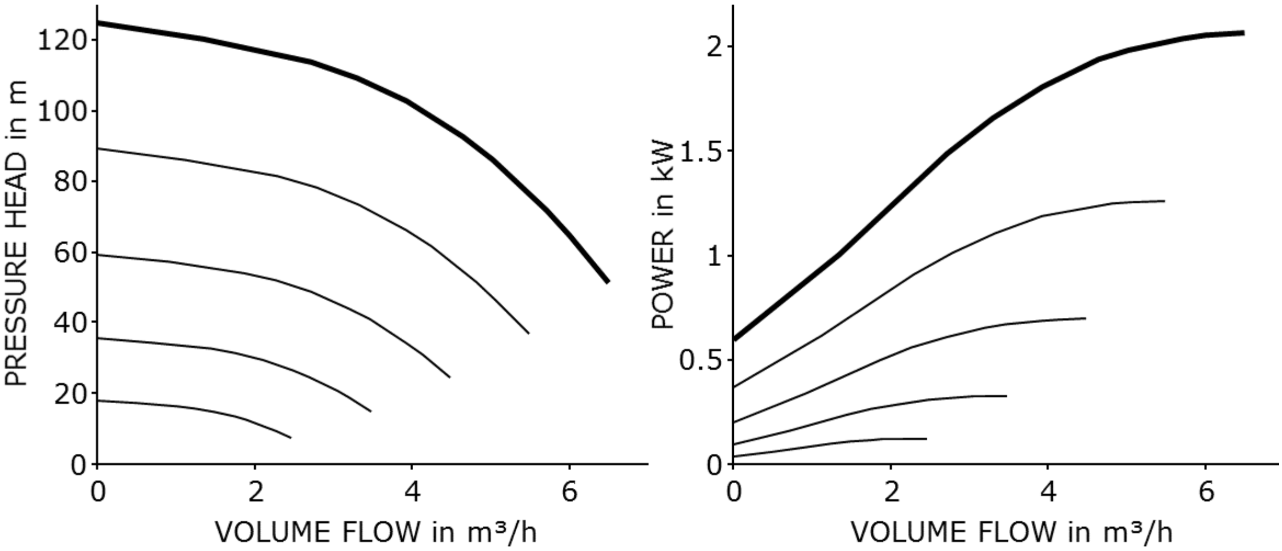

Figure 1: Characteristics of pump type A.

Table 1: Measurement points for pump type A.

FLOW RATE Q_{ref} in $m^3 h^{-1}$	PRESSURE HEAD H_{ref} in mWC	POWER INPUT P_{ref} in kW
0.0000	124.87	0.5981
1.3421	120.31	1.0031
2.7176	114.00	1.4878
3.2971	109.33	1.6588
3.9364	102.76	1.8072
4.6499	92.78	1.9402
5.0308	86.19	1.9832
5.7083	72.11	2.0403
6.0010	64.99	2.0567
6.5000	51.49	2.0649

Many marketwide available pumps are able to alter their rotational speed in the interval $[n_{min}, n_{max}]$. To calculate the characteristics for altering rotational speed, the affinity laws hold:

$$Q(n) = \left(\frac{n}{n_{ref}}\right) Q_{ref}, \tag{1}$$

$$\Delta H(n) = \left(\frac{n}{n_{ref}}\right)^2 \Delta H_{ref}, \tag{2}$$

$$P(n) = \left(\frac{n}{n_{ref}}\right)^3 P_{ref} \frac{1-\left(1-\eta_{opt}\right)\left(\frac{n}{n_{ref}}\right)^{-0.1}}{\eta_{opt}}. \tag{3}$$

Eq. (1) and (2) are the well known affinity laws. The affinity law for the power characteristics is expanded by an empirical correction term in order to account for the efficiency decrease due to decreasing rotational speed, where η_{opt} stands for the maximum hydraulic efficiency at reference speed. The basis of eq. (3) is a scaling law taking frictional laws into account [4]. Usually, but not necessarily, the maximal rotational speed of a pump is used as reference value.

Calculation of the Energy Efficiency Index

The EEI is the ratio of the average power input of the pump $P_{1,avg}$ to the reference power $P_{1,ref}$ of the booster station,

$$EEI := \frac{P_{1,avg}}{P_{1,ref}}. \tag{4}$$

The reference power is dictated by the maximal hydraulic power of the pump and equal for booster stations of the same nominal hydraulic data. This reference value is independent from the control and use of the booster station. Detailed information on the preliminary calculation of the reference power are provided by Taubert et al. [5].

To determine the average power input a load profile consisting of ten load points for the booster station is used [6]. All load points refer to the point of maximal hydraulic power, which defines the volume flow $Q_{100\,\%}$ and pressure head $\Delta H_{100\%}$. Table *2* shows the volume flow Q_l, pressure head ΔH_l and time portion t_l for the load profile with load cases $l = 1..10$.

Table 2: Load cases for the EEI of a booster station as suggested by Hirschberg [6].

LOAD CASE	1	2	3	4	5	6	7	8	9	10
Q_l in % of $Q_{100\,\%}$	10	20	30	40	50	60	70	80	90	100
ΔH_lin % of $\Delta H_{100\%}$	77.5	80	82.5	85	87.5	90	92.5	95	97.5	100
t_l in %	6	21	26	19	12	6	4	3	2	1

The resulting average power input is the time portion weighted average of the power input P_l in any load point:

$$P_{1,avg} := \sum_{l=1}^{10} t_l P_l. \tag{5}$$

The power input for the load points depends on the operation of the booster station: The degrees of freedom for the control of the booster station are the number of active pumps and the rotational speed of any active pump. The pressure difference over all active pumps must be equal. Different control strategies may result in different power inputs and thus different EEIs for the same machine.

3. Mathematical Model

While the rotational speed of a pump is controlled continuously within the technical available range, the number of activated pumps is a discrete decision. We present a purely mathematical approach which enables us to calculate the best attainable EEI for the machines data. Thus we gain a reference value for the applied operational strategy. The mathematical model consists of the pump characteristics, the flow model, and the objective for the optimization. Multiple use of the same model for all points of the load profile results in the EEI for the considered machine. Furthermore, applying the model for discrete points of the whole operating range leads to the optimal operational strategy for the booster station.

Pump Characteristics

From the characteristics measurement we gain several discrete data points for the operation of the pump at one specific rotational speed. Thus, we have to estimate values in between these points and to expand the data to create the field of operation. In general, we have two different possibilities for this estimation: A fitting function or piecewise linear interpolation. In this paper, we use the fitting fuction.

For the pump in our study, we used a 3rd degree polynomial to model the pressure characteristics and a 4th degree polynomial to model the power characteristics. The polynomial coefficients are given in Table *3*:

$$\Delta H_{ref}\left(Q_{ref}\right) = a_H Q_{ref}^3 + b_H Q_{ref}^2 + c_H Q_{ref} + d_H, \tag{6}$$

$$P_{ref}\left(Q_{ref}\right) = a_P Q_{ref}^4 + b_P Q_{ref}^3 + c_P Q_{ref}^2 + d_P Q_{ref} + e_P. \tag{7}$$

Table 3: Polynomial coefficients for pump type A.

COEFFICIENT	VALUE	COEFFICIENT	VALUE
a_H in mWC h^3 m^{-9}	-0.2448	a_P in kW h^4 m^{-12}	0.001357
b_H in mWC h^2 m^{-6}	0.3421	b_P in kW h^3 m^{-9}	-0.02259
c_H in mWC h m^{-3}	-3.197	c_P in kW h^2 m^{-6}	0.09047
d_H in mWC	124.9	d_P in kW h^1 m^{-3}	0.2196
		e_P in kW	0.59811

For altering the rotational speed the scaling laws (equations (2) and (3)) hold. By using additionally equation (1) we derive:

$$\Delta H(Q,n) = \left(\frac{n_{ref}}{n}\right) a_H Q^3 + b_H Q^2 + \left(\frac{n}{n_{ref}}\right) c_H Q + \left(\frac{n}{n_{ref}}\right)^2 d_H, \tag{8}$$

$$\begin{gathered} P(Q,n) = \\ = \left[\left(\frac{n_{ref}}{n}\right) a_P Q^4 + b_P Q^3 + \left(\frac{n}{n_{ref}}\right) c_P Q^2 + \left(\frac{n}{n_{ref}}\right)^2 d_P Q + \left(\frac{n}{n_{ref}}\right)^3 e_P\right] \cdot \\ \cdot \frac{1 - \left(1 - \eta_{opt}\right)\left(\frac{n}{n_{ref}}\right)^{-0.1}}{\eta_{opt}}. \end{gathered} \tag{9}$$

Flow Model

We model a fluid system as a mathematical flow graph $G(V,E)$. The edges E of the graph represent technical components, in this particular case the pump types K, or generic connections. The vertices V connect the edges [7]. Each edge transfers a volume flow. In each vertex the pressure is calculated. For every pump edge, a power input and the rotational speed is calculated.

The rotational speed has to be the same for all active pumps of one type [8]. Thus, each pump type is represented as one edge of the graph. We expand the variables from the section "Pump Characteristics" by in index l for any load case and i to account for the different pump types. We introduce an integer variable $x_{l,i}$ for each pump type that represents the number of active pumps for this pump type and modify the requirement for the load case:

$$Q_l = \sum_{\{i \ in \ K\}} x_{l,i} Q_{l,i}. \tag{10}$$

Due to the parallel arrangement, the volume flow for all pumps of one type equals the volume flow for a single pump multiplied by $x_{l,i}$. The required pressure head of the booster following from the given input and output pressure, is not influenced by the variable $x_{l,i}$ and still accounts for all pumps. Parallel pumps must have the same pressure head $\Delta H_{l,i}$. $H_{l,i,input}$ and $H_{l,i,output}$ represent the pressure at the input and output of the pump.

$$\forall i \in K \colon H_{l,i,input} + \Delta H_{l,i} = H_{l,i,output}. \tag{11}$$

If the range of $x_{l,i}$ includes 0, the pump type can be deactivated. The pressure head and the rotational speed, following from the pump characteristics model, will still be calculated, but the power input of the pump vanishes.

The correlation of the power input for multiple pumps is the same as of the volume flow and also uses variable $x_{l,i}$.

$$P_l = \sum_{\{i \ in \ K\}} x_{l,i} P_{l,i} \tag{12}$$

Objective

The objective for the optimization program is to minimize the total power input of all pumps for the given duty point.

$$min \ P_l \tag{13}$$

Application to large number of load cases

In order to quickly gain solutions for optimization problems of booster related tasks, we created an optimization framework within the AIMMS software system. The parameters for the optimization, such as different pump types as well as their characteristics, are given as user input. From the user input the flow graph is generated. The resulting optimization program is passed to a solver. The described optimization program finds an optimal control solution for one duty point. In order to find the maximal attainable EEI for a booster station or control guidelines for the whole field of operation, we have to run the optimization model many times with varying load cases.

4. Results

The best attainable EEI

For pump type A the point of maximal hydraulic power is $Q_{100\,\%} = 5.0308$ m^{3}h^{-1} and $H_{100\%} = 86.19$ mWC, the reference power is $P_{1,ref} = 6.27$ kW. To calculate the minimal average power input, we use the optimization program ten times with different load cases. Table *4* shows the optimal power input P_l, the number of active pumps $x_{l,A}$, and the rotational speed $n_{l,A}$ for these pumps. Hence, the best attainable EEI for this example is 0.31, no matter which operational strategy is applied for the use of the booster station.

Table 4: Result for the optimal EEI of a booster station consisting of three pumps of type A.

LOAD CASE	1	2	3	4	5	6	7	8	9	10
$P_l$in kW	0.54	0.95	1.48	2.03	2.57	3.20	3.80	4.44	5.16	5.95
$x_{l,A}$	1	1	1	2	2	2	3	3	3	3
$n_{l,A}$ in 10 rpm	218	235	262	241	254	270	256	267	278	290

Operational Strategy for Booster Stations with one Pump Type

Figure *2* shows the corresponding optimal operational strategy for the booster station. We can identify two reasons for the switch of a pump: (1) For high pressure and increasing volume flow the controller increases the rotational speed of the active pumps. Once the maximum is reached, an additional pump is switched on. (2) For low pressure and increasing volume flow we find an efficiency argument: An additional pump is switched on, because it reduces the total power input, even though more pumps are working than necessary.

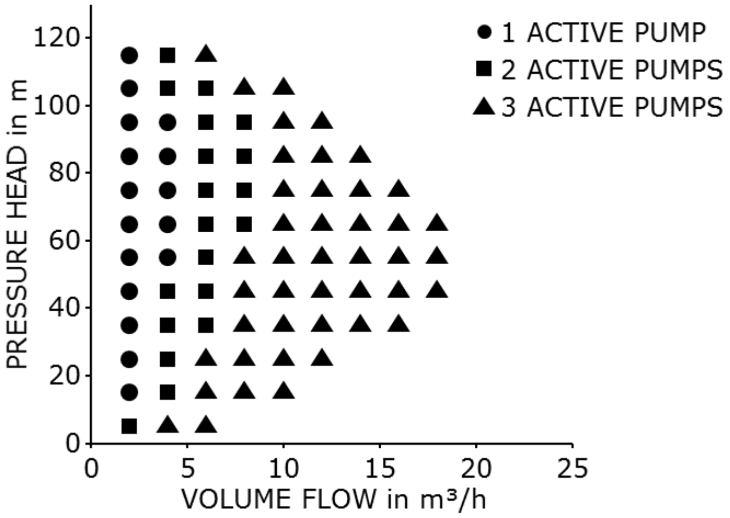

Figure 2: Optimization result for a booster station of three pumps of type A.

Operational Strategy for Booster Stations with two Pump Types

The second considered optimization problem is a booster station with three pumps of type A and one pump of different type (type B). Table *5* shows the measurement points for pump type B. In this case, we have to deal with a booster station consisting of non-identical pumps. Therefore we cannot assume the rotational speeds to be equal. Hence, another degree of freedom is added to the optimization problem. The maximal hydraulic efficiency of the pump type B is worse than pump type A, but the nominal volume flow is less. So we expect, that the additional pump reduces the overall power input of the booster station in some operational points. Figure *3* shows the results for the number of active pumps. The additional pump allows further energy savings in particular ranges of the operational field, but further complicates the operational strategy of the booster station: The additional pump can only be used in some operational points and thus has to be switched on and off for constant pressure and increasing volume flow.

Table 5: Measurement points for pump type B.

FLOW RATE Q_{ref} in m³h⁻¹	PRESSURE HEAD H_{ref} in mWC	POWER INPUT P_{ref} in kW
0.0000	123.86	0.4597
0.6000	120.80	0.6289
1.2215	114.09	0.7930
1.6223	107.30	0.9069
1.8152	103.55	0.9673
1.9219	101.30	0.9887
2.0529	86.11	1.0242
2.4428	72.11	1.1013
2.6902	75.95	1.1334
3.0266	59.32	1.1545

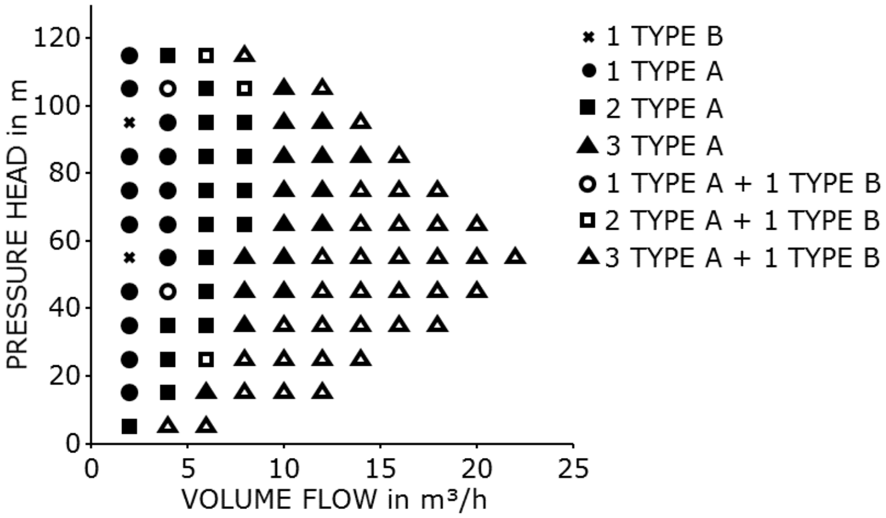

Figure 3: Optimization result for a booster station of three pumps of type A and one pump of type B.

5. Conclusion

In this paper we showed an approach to calculate the best attainable EEI for a booster station for the single machines date. Our purely mathematical approach uses an optimization program to calculate the minimal power input for any operational point. Our result represents the technically attainable minimum for the EEI and enables designers and operators to rate the operational strategy of a specific booster station.

Furthermore the approach allows to propose the optimal operational strategy for a booster station of a given set of pumps. We made an improvement suggestion for the booster station by adding an additional pump of a different type. The additional pump allows for further reduction of the energy input, but leads to a more complicated operational strategy.

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References

[1] Pelz, P. F.: 250 Jahre Energienutzung: Algorithmen übernehmen Synthese, Planung und Betrieb von Energiesystemen, Fachvortrag anlässlich der Ehrenpromotion von Hans-Ulrich Banzhaf, Universität Siegen, 2014.

[2] Bidstrup, N.; Teepe, M.; Berge, G.; Ludwig, G.: Extended Product Approach for Pumps, Draft Version, Europump, 2014.

[3] Pelz, P.; Lorenz, U.: As good as can be! TOR plans the energetically optimal fluid system. ErP Special Issue, Delta p, 2013.

[4] Pfleiderer, C.: Die Kreiselpumpe für Flüssigkeiten und Gase, Wasserpumpen, Ventilatoren, Turbogebläse, Turbokompressoren, Springer-Verlag, 1961.

[5] Taubert, P.; Stoffel, B.; Ludwig, G.; Pelz, P.F.: Development of a Standardized Approach to Assess the Energy Efficiency of Booster Pump Units in the Sense of an Extended Product, Proceedings of Pump Users International Forum 2016, 2016.

[6] Hirschberg, R: Lastprofil und Regelkurve zur energetischen Bewertung von Druckerhöhungsanlagen (DEA), HLH Lüftung/Klima, Heizung/Sanitär, Gebäudetechnik, ISSN: 1436-5103, Bd.65, Nr.10, S.68-73, 2014.

[7] Pöttgen, P.; Ederer, T.; Altherr, L. C.; Pelz, P. F.: Developing a Control Strategy for Booster Stations under Uncertain Load. Applied Mechanics and Materials, 807 pp. 241-246. ISSN 1660-9336, 2015.

[8] Groß, T. F.; Pöttgen, P. F.; Pelz, P. F.: An Analytical Approach for the Optimal Operation of Pumps in Booster Systems, http://arxiv.org/abs/1606.05423.

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