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Prediction of the Effect of Impeller Trimming on the Hydraulic Performance of Low Specific-Speed Centrifugal Pumps

The effect of trimming of radial impellers on the hydraulic performance of low specific-speed centrifugal pumps is studied. Prediction methods from literature, together with a new prediction method that is based on the simplified description of the flow field in the impeller, are used to quantify the effect of trimming on the hydraulic performance. The predictions by these methods are compared to measured effects of trimming on the hydraulic performance for an extensive set of pumps for flow rates in the range of 80% to 110% of the best efficiency point. Of the considered methods, the new prediction method is more accurate (even for a large impeller trim of 12%) than the considered methods from literature. The new method generally overestimates the reduction in the pump head after trimming, and hence results less often in impeller trims that are too large when the method is used to determine the amount of trimming that is necessary in order to attain a specified head. [DOI: 10.1115/1.4039251]

1 Introduction

When the operating conditions of a centrifugal pump change, an existing pump may be reworked in order to match its performance to the new operating conditions. Additionally, it may be advantageous to use a pump design (with an available casting pattern) to cover a certain performance range and to use rework to yield adjustments to its performance. Rework of impellers is based on modifications to the impeller geometry. Rework to the leading edge is discussed in Ref. [1]. The focus in the current study is on rework of the trailing edge for low specific-speed centrifugal pumps.

Many types of rework of the trailing edge are employed. These include the following: (1) overfiling (removing material at the pressure side of the blades near the trailing edge), (2) underfiling (removing material at the suction side of the blades near the trailing edge), (3) squaring the trailing edge, and (4) impeller *trimming* (or impeller cut-back), i.e., the reduction of the outer diameter of the impeller (by machining the blade at the trailing edge). Trimming is usually performed at a constant radius (in axial direction), but an oblique trim is possible as well. With oblique trimming, the impeller diameter is trimmed under an angle (between 5 deg and 15 deg) with respect to the axial direction.

The qualitative effects of these types of rework on the hydraulic performance characteristics are well described in Refs. [1] and [2]. The most often used type of rework is trimming, as it is relatively easy and inexpensive to perform.

Trimming of the impeller blades leads to a lower impeller tip speed (at constant angular velocity), and hence to a lower head as follows from the Euler pump equation (see, for example, Ref. [3]). A reduced impeller diameter leads to higher blade loading and lower flow deflection. Hence, the impeller efficiency is expected to be lower after trimming [2]. After trimming, the gap between the trailing edge of the impeller and the volute tongue becomes larger, resulting in an increase in flow recirculation and in volute losses. Disk friction losses will be smaller after trimming. Leakage losses will also be smaller as the leakage flow is reduced after trimming, since the impeller head is smaller. After trimming, the best efficiency point (BEP) shifts to a lower flow rate [1]. Gülich [2] gives a correlation for the efficiency drop due to trimming. The decrease of efficiency after trimming has been observed experimentally, see for instance Refs. [2] and [4]. The effect of trimming on the hydraulic performance is illustrated in Fig. 1.

For volute pumps, the complete impeller (including the front and rear shroud) is generally cut back to a smaller diameter. For

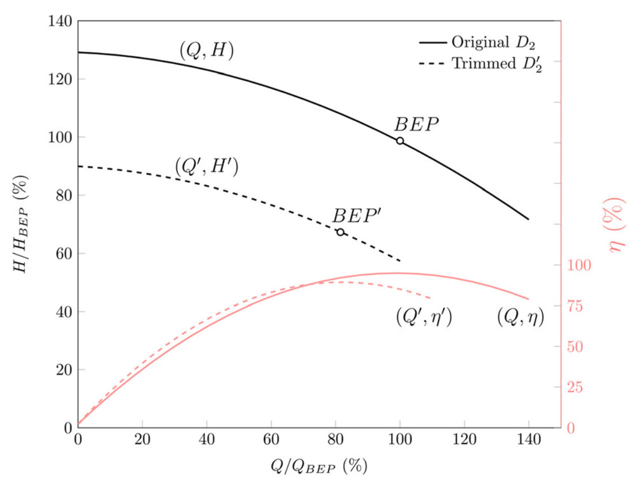

Fig. 1 Effect of trimming on the hydraulic performance. The head, flow rate, and efficiency of the untrimmed impeller are denoted by (H, Q, η), while the corresponding performance of the trimmed impeller is (H', Q', η'). The best efficiency points are denoted by BEP and BEP′.

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diffuser pumps, only the blades are trimmed and the shrouds remain the same, since the pump characteristics may become unstable if the shrouds of a diffuser pump are also trimmed [2]. As the shrouds of diffuser pumps are not trimmed, disk friction losses after trimming are higher for diffuser pumps than for volute pumps. Therefore, the efficiency loss due to trimming is larger for diffuser pumps than for volute pumps.

Barrio et al. [5] studied (numerically and experimentally) the effect of trimming on radial forces on the impeller at the blade passing frequency in a volute pump. It was found that the magnitude of these forces increases with increasing impeller trimming, especially at off-design conditions.

Results of computational fluid dynamics (CFD) simulations for the effects of various trimming methods of the trailing edge (such as, straight, triangle, and oblique trimming) on the hydraulic performance are given in Refs. [6] and [7]. Results of CFD simulations for the effect of rounding of the trailing edge of the impeller of a mixed-flow pump on the hydraulic characteristics are presented in Ref. [8]. Similar CFD-based simulations were reported in Ref. [9] for a low specific-speed centrifugal pump, who also considered the influence on unsteady pressure fluctuations.

Compressible CFD simulations have been performed by Swain and Engeda [10] to investigate the effect of modifications to the axial blade height (axial trimming, contrary to the radial trimming considered here) on the performance of an open (i.e., unshrouded) centrifugal compressor.

Yang et al. [11] studied the effect of trimming on the performance of pumps in pump and pump-as-turbine modes of operation. They performed experiments as well as CFD simulations of the flow in the pumps, for various impeller diameters.

Impeller trimming is generally performed in small steps because of the uncertainty in predicting the effect of trimming on the hydraulic performance, as it is easier to remove material than to add material. Consequently, the pump may need to be tested and reworked multiple times, which results in increased costs and lead time. These added costs can be reduced by using a method that predicts the effects of trimming on the hydraulic performance of the pump more accurately. Various methods to predict the effect of impeller trimming on the hydraulic performance are considered here.

Requirements for such prediction methods are as follows:

(1) Reference measurements for the original, untrimmed impeller are available.
(2) Accurate around the best efficiency point.
(3) Do not lead to trims that are too large (overtrimming).
(4) Simplicity in use.

This last requirement precludes the use of CFD-based methods (for example, Refs. [6–9]) in many situations, as these are generally (too) complex and time-consuming in comparison to their accuracy. With respect to the accuracy of prediction methods, performance tolerances are important. The API standard 610 [12] specifies tolerances for the head of $\pm 3\%$.

Although impeller trimming is frequently performed, limited information is available in the open literature on the accuracy of prediction methods. Therefore, a (wide) range of methods from literature is considered here, together with a new method developed here. The predictions by the various methods are compared with extensive data for centrifugal pumps with a range of specific speeds (0.22–0.59; the specific speed is defined in Eq. (5)).

The outline of this study is as follows: Various prediction methods are described in Sec. 2. These range from methods from literature that only take into account changes in outlet diameters (see Sec. 2.1), to a method from literature that also takes into account the inlet diameter (see Sec. 2.2) and to a new method that is based on a simplified description of the flow field (see Sec. 2.3). The considered centrifugal pumps and the measurements of their hydraulic performance are described in Sec. 3. A comparison of the results of these prediction methods with the measured performance after trimming is given in Sec. 4.

2 Prediction Methods

The description of the prediction methods is given for the case where the angular velocity Ω of the impeller remains unchanged after trimming. The (volumetric) flow rate is denoted by Q. The diameter of the original, untrimmed impeller is denoted by D_2, while that of the trimmed impeller is D_2'. The head of the untrimmed pump is denoted by H, while that of the trimmed impeller is H'. The performance point (Q, H) of the untrimmed impeller corresponds to the performance point (Q', H') of the trimmed impeller. Scaling methods provide relationships between the performance points (Q, H) and (Q', H').

Changes in the efficiency due to trimming are not considered here, as these are of secondary importance and are difficult to predict [2].

2.1 Scaling Methods.

The scaling methods are based on the assumption that flow angles do not change with trimming, i.e., the ratio of the circumferential component $c_{\theta 2}$ to the meridional component c_{m2} of the absolute velocity at the trailing edge of the impeller $c_{\theta 2}/c_{m2}$ is constant. It follows from the Euler pump equation (see, for example, Ref. [3]) that the head H scales with the product of the tip speed u_2 and the circumferential velocity at the trailing edge $c_{\theta 2}$. As $u_2 \propto D_2$ and $c_{\theta 2} \propto D_2$, it follows that the head $H \propto D_2^2$. This proportionality is common to all scaling methods. The meridional velocity c_{m2} is determined by the flow rate Q and the through-flow area A_2 at the trailing edge by $c_{m2} = Q/A_2$. This through-flow area is given by $A_2 = \pi D_2 b_2$ (for low-specific speed, radial impellers and neglecting the blockage effect due to impeller blade thickness), where b_2 is the (axial) width of the impeller (as indicated in Fig. 2(*a*)). Different scaling methods are obtained, depending on the adopted assumption for the scaling of the through-flow area A_2 with the diameter D_2, as is shown below.

With geometrical scaling [1], it is assumed that the impeller is geometrically scaled in all dimensions ($b_2 \propto D_2$, and hence $A_2 \propto D_2^2$). This results in the classical scaling relations (see, for example, Ref. [3]) for the influence of the impeller diameter on the hydraulic performance for geometrically similar impellers

$$\text{geometrical scaling}: \quad \frac{Q'}{Q} = \left(\frac{D_2'}{D_2}\right)^3 \quad \frac{H'}{H} = \left(\frac{D_2'}{D_2}\right)^2 \tag{1}$$

According to Ref. [1], geometrical scaling can be employed for small to moderate changes in the impeller diameter ($D_2'/D_2 \geq 0.9$).

With constant-width scaling, it is assumed that the width b_2 of the impeller does not change ($b_2 = \text{const}$). This leads to the scaling relations

$$\text{constant-width scaling}: \quad \frac{Q'}{Q} = \left(\frac{D_2'}{D_2}\right)^2 \quad \frac{H'}{H} = \left(\frac{D_2'}{D_2}\right)^2 \tag{2}$$

Sulzer Pumps [13] recommends this method when $D_2'/D_2 \geq 0.94$. This method was also used in Ref. [4] to scale the performance after trimming for their impeller with constant width b_2 and in Ref. [14] for impellers where the front and back shrouds were not trimmed.

With constant-area scaling, it is assumed that the through-flow area of the impeller does not change ($A_2 = \text{const}$). This leads to the scaling relations [15,16]

$$\text{constant-area scaling}: \quad \frac{Q'}{Q} = \frac{D_2'}{D_2} \quad \frac{H'}{H} = \left(\frac{D_2'}{D_2}\right)^2 \tag{3}$$

An empirical correction to the constant-area scaling method in Eq. (3) has been described in Refs. [15] and [16]. With specified diameter D_2, head H, and head after trimming H', the scaling relation Eq. (3) provides an estimate of the required trimmed diameter D_2'. Based on experimental data, it was noted that this estimate

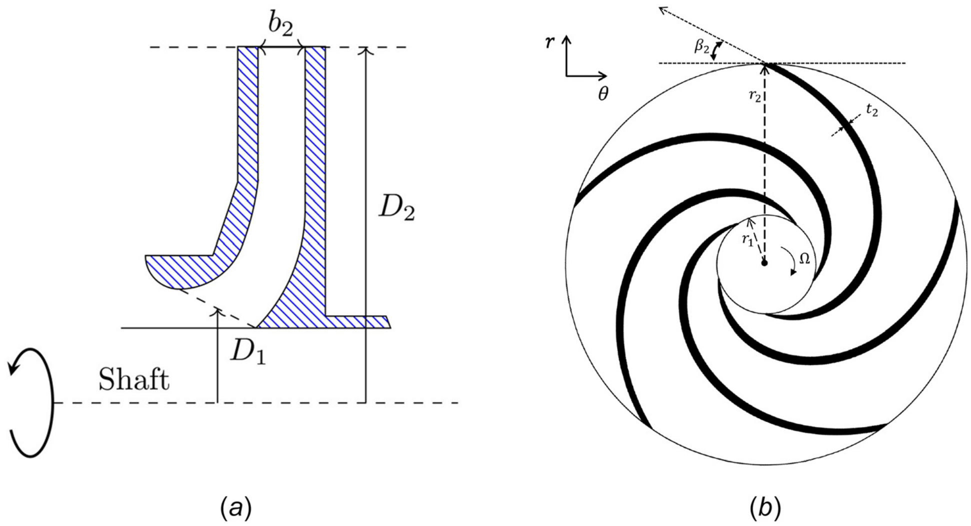

Fig. 2 **(*a*) Meridional geometry with the definition of the inlet and outlet diameters, D_1 and D_2, respectively, and the axial width b_2 and (*b*) planar geometry of the impeller with the definition of the blade angle β_2 and the blade thickness t_2; here, the number of blades $Z = 5$**

leads to a trim that is too large. Therefore, it has been suggested to employ a corrected diameter $D'_{2,\mathrm{corr}}$ given by $1 - D'_{2,\mathrm{corr}}/D_2 = \tau(1 - D'_2/D_2)$ (with $\tau \cong 5/6$) as given graphically in Ref. [15].

This leads to the corrected constant-area scaling relation for the head

$$\text{corrected constant-area head-scaling:}\quad \frac{H'}{H} = \left[1 - \frac{1}{\tau}\left(1 - \frac{D'_2}{D_2}\right)\right]^2 \tag{4}$$

Note that this corrected head-scaling method yields a trimmed diameter D'_2 that is larger than that obtained from the "uncorrected" head scaling, Eq. (3). This means that this method will less often lead to an impeller trim that is too large.

Trimming may change the specific speed according to the three scaling relations in Eqs. (1)–(3). The (dimensionless) specific speed Ω_s with the untrimmed impeller is defined by

$$\Omega_s = \Omega \frac{Q^{1/2}}{(gH)^{3/4}} \tag{5}$$

where g is the gravitational acceleration and all quantities correspond to those at the best efficiency point. The specific speed with the trimmed impeller, Ω'_s, is defined analogously. It follows from Eq. (1) that according to the geometrical scaling method $\Omega'_s/\Omega = 1$, while according to the constant-width scaling method in Eq. (2) $\Omega'_s/\Omega = (D'_2/D_2)^{-1/2}$. Finally, according to the constant-area scaling method in Eq. (3), $\Omega'_s/\Omega = (D'_2/D_2)^{-1}$.

2.2 Inlet-to-Outlet Scaling. The inlet-to-outlet scaling method is given in Refs. [2] and [17]. In this method, the inlet diameter D_1 is taken into account in the scaling relations between the performance points (Q, H) and (Q', H')

$$\text{inlet-to-outlet scaling:}\quad \frac{Q'}{Q} = \sqrt{\frac{D'^2_2 - D_1^2}{D_2^2 - D_1^2}} \quad \frac{H'}{H} = \frac{D'^2_2 - D_1^2}{D_2^2 - D_1^2} \tag{6}$$

Here, D_1 is a representative diameter of the leading edge, in between the hub and the shroud (see also Fig. 2(*a*)). The inlet diameter D_1 does not change with trimming of the trailing edge of the impeller blades. No physical motivation for this method is provided. Note that for small inlet diameters D_1, $D_1 \to 0$, this method reduces to the constant-area scaling method, Eq. (3).

With the inlet-to-outlet scaling method, the factors by which the flow rate and the head are scaled are smaller than with the constant-area scaling method, Eq. (3).

2.3 Difference Method. The novel *difference method* that is described here is based on the availability of geometrical information on the impeller (often as information from computer-aided design-systems or as drawings) and of measurements of the hydraulic performance of the untrimmed pump. No new empirical parameters are required.

Employing this geometrical information, the theoretical (ideal) head H_{th} can be determined, based on a simplified description of the flow through the impeller, where the flow field is assumed to be one-dimensional (1D) (i.e., uniform from blade to blade and in spanwise direction). Deviations from this one-dimensional flow are accounted for through a slip factor from literature that accounts for the deviation of the average flow angle from the geometrical blade angle β_2. The slip factor γ is defined here by $1 - \gamma = c_{\theta s}/u_2$, where $c_{\theta s}$ is the slip velocity (see, for instance, Refs. [2] and [3]).

The Euler pump equation (see, for example, Ref. [3]) for the theoretical head H_{th} is

$$gH_{\mathrm{th}} = \Omega\frac{D_2}{2}\left[\gamma\Omega\frac{D_2}{2} - \frac{Q}{\tan\beta_2 A_2}\right] \quad A_2 = \left(\pi D_2 - Z\frac{t_2}{\sin\beta_2}\right)b_2 \tag{7}$$

Here, Ω is the angular velocity (in rad/s), Q is the flow rate, γ is the slip factor, and β_2 is the blade angle (with respect to the circumferential direction) at the trailing edge, Z is the number of blades on the impeller, t_2 is the blade thickness at the trailing edge, and b_2 is the axial width at the trailing edge. These geometrical quantities are illustrated in Fig. 2(*b*). Note that the expression for the through-flow area A_2 in Eq. (7) accounts for blockage of the flow by the blades. The slip factor by Busemann [18] has been employed (see also Ref. [19]).

With expression Eq. (7), the theoretical head of the untrimmed impeller and of trimmed impeller, $H_{\mathrm{th}}(Q)$ and $H'_{\mathrm{th}}(Q)$, respectively, can be determined.

Deviations between this theoretical head $H_{\mathrm{th}}(Q)$ and the measured head $H_{\mathrm{test}}(Q)$ of the untrimmed impeller are the result of leakage losses and of hydraulic losses in the impeller and the volute. Disk friction losses and mechanical losses result in a higher pump torque and hence affect the overall efficiency. After

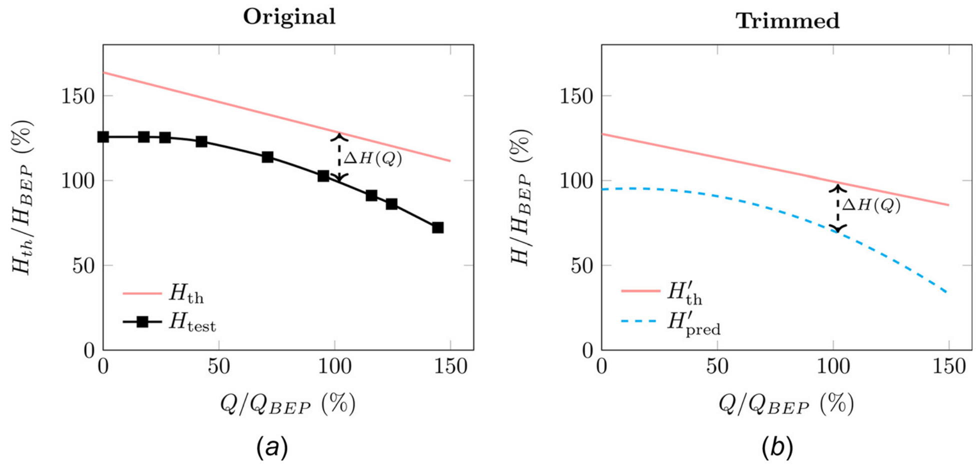

Fig. 3 Graphical representation of the difference method. Left: Theoretical head curve $H_{\rm th}(Q)$ according to Eq. (7) and measured head curve $H_{\rm test}(Q)$; both with the original, untrimmed impeller. Right: Theoretical head curve $H'_{\rm th}(Q)$ with the trimmed impeller and prediction for the head curve $H'_{\rm pred}(Q)$ of the trimmed impeller; $\Delta H(Q)$ is defined in Eq. (8).

trimming, hydraulic losses will *increase* due to a higher blade loading and a larger gap between the trailing edge of the impeller and the volute tongue. On the other hand, leakage losses will *decrease* as a result of the reduction in head after trimming. The separate quantification of these counteracting effects (in a relatively simple manner) would require models that would contain (many) empirical parameters. In order to propose a prediction method that does not require additional empirical information, it is assumed that the combined effect of (counteracting) hydraulic and leakage losses does not affect the losses after trimming.

Hence, it is assumed that losses, characterized by the difference $H_{\rm th}(Q)$–$H_{\rm test}(Q)$ between the theoretical head $H_{\rm th}(Q)$ and the actual head as determined in a test $H_{\rm test}(Q)$, remain unchanged after trimming. The predicted head after trimming $H'_{\rm pred}(Q)$ according to the difference method is then given by

$$H'_{\rm pred}(Q) = H'_{\rm th}(Q) - \Delta H(Q) \quad \Delta H(Q) = H_{\rm th}(Q) - H_{\rm test}(Q) \tag{8}$$

This procedure is graphically represented in Fig. 3.

The difference method is more detailed than the scaling methods from Secs. 2.1 and 2.2, as geometrical characteristics of the impeller (such as the number of blades Z, blade angle β_2, blade width b_2, and blade thickness t_2 at the trailing edge) of both untrimmed and trimmed impeller are taken into account in the one-dimensional description of the flow that ultimately determines the hydraulic performance. In addition, losses $\Delta H(Q)$ that represent the difference between the theoretical head curve and the measured head curve (both with the untrimmed impeller) are accounted for, without having to introduce empirical parameters.

3 Pumps and Measurements

Pump test data have been collected for three different types of multi-stage pumps that have been tested by Flowserve during the period 2011–2015. These data have been obtained from so-called shop tests that have been performed in accordance with the Hydraulic Institute Test Standard [20]. These performance tests have been conducted with water at various flow rates in a semi-closed test loop. In order to control the water temperature, the warmed up water has been partly extracted from the loop and replaced by cooled water. The measurements have been done according to the API 610 11th edition standard [12]. The temperature and pressure of the water have been measured in the pipes before entering and after exiting the pump using calibrated temperature and pressure gauges. The inaccuracy of the calibrated measuring devices is small: $\pm 0.2\%$.

The collected test data have been filtered for data in which the impeller has only been trimmed (no other rework has been performed) and for which the hydraulic performance characteristics of both untrimmed and trimmed impellers were available. In order to use the difference method from Sec. 2.3, geometrical data are also required. Twelve cases were found, whose primary data are given in Table 1 in the two columns on the left. These pumps are low specific-speed, high-pressure, horizontal multistage between bearing pumps. Pumps 1, 2, 3, 6, 9, and 11 from this table are BB3 pumps and the others pumps are BB5 pumps [12]. All these pumps have closed (i.e., shrouded) impellers of which the first stage impeller has a special design for better cavitation performance. This first stage impeller has a diameter that differs from that of all other impellers (so-called series impellers). Most pumps have one type of series impellers, but some of the listed pumps have two geometrically different types of series impellers but with the same outer diameter. Using different types of series impellers allows for tuning the steepness of the head curve.

Table 1 Accuracy of prediction methods. Left two columns: reduction in impeller diameter (Trim %) and specific speed N_s (defined in Eq. (5); the specific speed is based on the head and the flow rate at the best efficiency point with the untrimmed impeller)

Trim %	N_s	Eq. (1)	Eq. (2)	Eq. (4)	Eq. (6)	Eq. (8)
1.40	0.50	0.21	−0.47	−0.67	0.27	0.28
1.75	0.49	0.41	−0.75	−1.26	0.01	−0.02
2.05	0.49	0.31	−0.61	−0.85	−0.31	0.08
2.10	0.34	−1.00	−2.33	−2.90	−2.12	−1.70
2.46	0.27	1.35	−0.28	−0.99	0.10	−0.71
2.79	0.58	1.65	−0.38	−1.33	−2.16	1.94
3.08	0.32	2.50	0.86	0.29	1.86	2.21
3.24	0.28	1.09	−0.58	−1.08	0.30	0.65
3.32	0.59	0.07	−1.89	−2.55	0.40	1.68
3.64	0.22	1.08	−0.40	−0.62	−0.97	0.05
3.65	0.56	2.25	0.01	−0.84	1.71	1.46
12.23	0.26	11.48	2.21	−0.41	−1.27	1.28

Note: Other columns: error (in %) according to Eq. (10) in the prediction methods. Cells in gray indicate negative errors, leading to overtrimming. Equation (1): geometrical scaling method; Eq. (2): constant-width scaling method; Eq. (4): constant-area scaling method with correction; Eq. (6): inlet-to-outlet scaling method; Eq. (8): difference method.

The prediction methods presented in Sec. 2 have been developed for a single stage, while the considered pumps are multistage pumps. To calculate the average head of a stage from experimental data for a multistage pump, the effect of the first-stage impeller is excluded as the first stage of the considered pumps may have slightly different geometrical characteristics. The impeller diameter of the first stage is denoted by D_{2f}, while that of the other stages is D_2. The head of a stage H is determined from the head H_{multi} of the multistage pump by

$$H = \frac{H_{\text{multi}}}{N_{\text{stage}} - 1 + \left(\frac{D_{2f}}{D_2}\right)^2} \tag{9}$$

where N_{stage} is the number of stages and the term involving the diameter ratio D_{2f}/D_2 accounts for the difference in contribution to the head of the first stage in comparison to the contribution to the head of the other series stages. The (relative) contribution of the first stage scales with $(D_{2f}/D_2)^2$.

This method of determining the head per stage H from the head H_{multi} of the multistage pump has been verified, using data for a multistage pump for which the head of the first stage, H_f, has been measured separately. For this case, the difference between the (stage) head H according Eq. (9) and that obtained from the measured pump head H_{multi} and first-stage head H_f according to $H = (H_{\text{multi}} - H_f)/(N_{\text{stage}} - 1)$ was smaller than 0.3% over the range of considered flow rates.

The Reynolds number $\text{Re} = \Omega D_2^2/\nu$ (with ν the kinematic viscosity of the fluid) of the untrimmed pumps ranged from 2.3×10^7 to 5.5×10^7.

4 Results

The methods from Sec. 2 give, using the measured head curve $H_{\text{test}}(Q)$ with the untrimmed impeller, a prediction for the head curve $H'_{\text{pred}}(Q)$ after trimming.

The accuracy of the prediction methods is evaluated by comparing the predicted head $H'_{\text{pred}}(Q)$ with the measured head $H'_{\text{test}}(Q)$, both for the trimmed impeller. The error in the prediction of the head, $H_{\text{err}}(Q)$, is defined by

$$H_{\text{err}}(Q) = \frac{H'_{\text{test}}(Q) - H'_{\text{pred}}(Q)}{H'_{\text{test}}(Q)} \tag{10}$$

When a prediction method is used to determine the amount of trimming that is required to achieve a specified head after trimming, then it is undesirable if the method overestimates the head. The required amount of trimming suggested by the prediction method is then too large, so *more* material is removed than necessary. On the other hand, if a prediction method underestimates the head, then too little material would be removed in order to achieve a specified head after trimming. As it is much easier to remove additional material (extra trimming) than to add material, underestimation of the head after trimming is considered to be less severe than an overestimation. Hence, positive errors of a prediction method are considered to be less severe than negative errors.

Examples of the predicted head curve after trimming, $H'_{\text{pred}}(Q)$, are shown in Figs. 4 and 5 (left), together with the measured head curves with the untrimmed impeller $H_{\text{test}}(Q)$ and with the trimmed impeller $H'_{\text{test}}(Q)$. The corresponding errors, $H_{\text{err}}(Q)$ as defined in Eq. (10), are shown in Figs. 4 and 5 (right). The results shown in Fig. 4 are for the geometrical scaling method of Eq. (1). This (inaccurate) method has been considered here, as the different curves are clearly discernible. For small flow rates Q the head is overpredicted (negative error H_{err}), while for high flow rates the head is significantly underpredicted (large positive error H_{err}). In addition, results are shown for the difference method of Eq. (10) in Fig. 5. For small flow rates, the error is large and positive, while around Q_{BEP} the error is small and positive.

Rather than showing results for all methods as done for the geometrical scaling method in Fig. 4 and for the difference method in Fig. 5, the accuracy of the methods is characterized as follows: The mean errors are calculated for each method, based on all measured points between 80% to 110% of the best efficiency point, since the rated point should be within this range according to the API standard 610 [12]. In Figs. 4 and 5, this range is indicated by the hatched rectangle. For the various prediction methods from Sec. 2, the mean errors according to Eq. (10) are listed in Table 1.

The geometrical scaling method, Eq. (1), predicts the performance after trimming with positive errors (i.e., trims that are not too large) for all cases except a single case where the error is slightly negative. The magnitude of the (positive) errors is larger than for the other methods, especially for the case with the large trim of 12.23% where the error is unacceptably large (11.48%). The geometrical scaling method is accurate (error smaller than 1.4%) for trims of up to 2.5%.

The predictions according to the constant-width scaling method, Eq. (2), show negative errors (of moderate magnitude; up to 2.3%) for most cases, leading to moderate overtrimming.

The predictions according to the constant-area scaling method with correction, Eq. (4), show negative errors (of moderate

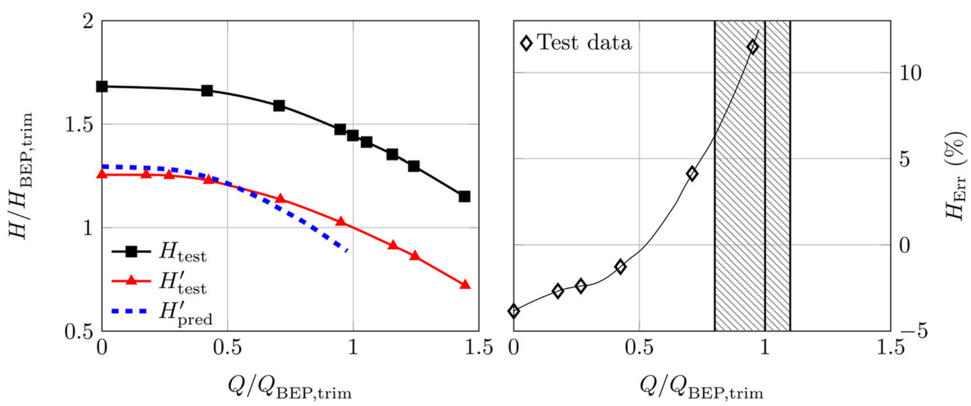

Fig. 4 Left: Comparison between the experimental head curve after trimming $H'_{\text{test}}(Q)$ and the predicted head curve after trimming $H'_{\text{pred}}(Q)$ according to the *geometrical scaling method*, Eq. (1). Also shown is the experimental head curve $H_{\text{test}}(Q)$ with the original, untrimmed impeller. Right: Error $H_{\text{err}}(Q)$ in the prediction method, as defined in Eq. (10). The hatched rectangle indicates the range of flow rates between 80% to 110% of the BEP. Results given here are for the pump with the large trim of 12.23% (see Table 1).

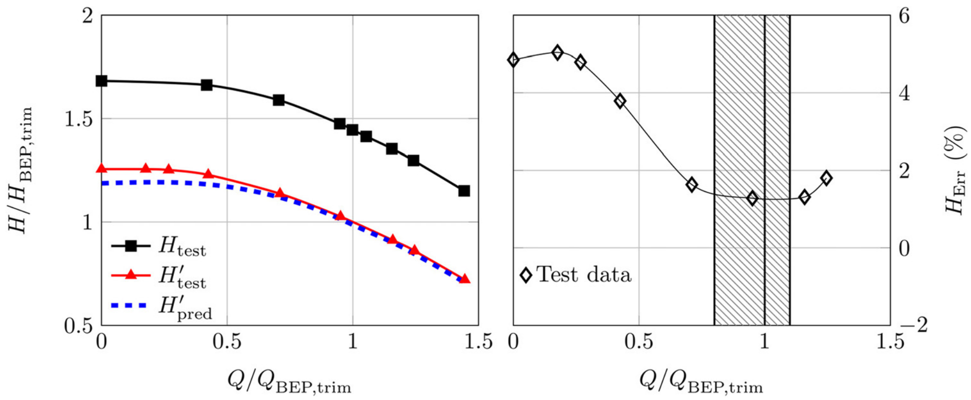

Fig. 5 Left: Comparison between the experimental head curve after trimming $H'_{\mathrm{test}}(Q)$ and the predicted head curve after trimming $H'_{\mathrm{pred}}(Q)$ according to the difference method, Eq. (8). Also shown is the experimental head curve $H_{\mathrm{test}}(Q)$ with the original, untrimmed impeller. Right: Error $H_{\mathrm{err}}(Q)$ in the prediction method, as defined in Eq. (10). The hatched rectangle indicates the range of flow rates between 80% to 110% of the BEP. Results given here are for the pump with the large trim of 12.23% (see Table 1).

magnitude; up to 2.9%), leading to moderate overtrimming. The constant-area scaling method with correction is (much) more accurate (by 1.2% on average) than the (plain) constant-area scaling method, Eq. (3) (results not shown).

The predictions according to the inlet-to-outlet scaling method, Eq. (6), show negative errors (of moderate magnitude; up to 2.2%), leading to moderate overtrimming, for about half of the considered pumps. In the other cases, the (positive) errors are fairly small.

The errors for the difference method, Eq. (8), are generally small and positive (i.e., leading to trims that are not too large). For the cases where the error is negative, the magnitude is generally also small.

Correlation coefficients have been determined between trimming percentage and error. Only for the geometrical scaling method, Eq. (1), and the constant-width scaling method, Eq. (2), a significant correlation (correlation coefficient larger than 0.6) has been observed. No significant correlation between specific speed, as defined in Eq. (5), and error has been noted for the considered methods.

4.1 More Detailed Prediction Methods. CFD-based methods can be used to investigate the effect of trimming on rework of the trailing edge (see, for example, Wu [6–9]). In a preliminary study, CFD simulations of the flow in a single impeller channel have been performed by Detert Oude Weme [21] with a commercial CFD-code in which the incompressible Reynolds-averaged Navier–Stokes equations have been solved, employing the SST-turbulence model. The pump with the large trim of 12.23% (see Table 1) has been considered. At the inlet, a uniform velocity is prescribed (based on the flow rate and without preswirl), while at the outlet a uniform pressure is prescribed. At the blades, the hub and the shroud the no-slip condition for the relative velocity is enforced. A second-order high-resolution scheme has been employed for the discretization of the governing equations using the finite volume method (with co-located variables). The solution of the discretized equations has been obtained iteratively, with a reduction in residuals by a factor of at least 10^{-5}. The average nondimensional distance y^+ [22] of the first nodes from the solid walls $y^+ \cong 24$. A grid convergence study has been performed for the untrimmed impeller at the Best Efficiency Point to verify that the obtained results are independent of the grid size. The finest grid contained about 930,000 nodes. As the volute and the leakage flow path were not considered, only hydraulic losses in the impeller were obtained, with a peak hydraulic efficiency $\eta_h \cong 96\%$.

The *difference* in head between untrimmed and trimmed impellers predicted by the CFD-based method was almost the same (within 1%) as the difference in head obtained with the difference method from Sec. 2.3. This study shows that such (time-consuming) CFD-based methods may not be more accurate than simpler methods, as considered in Sec. 2, for predicting the effect of trimming on the hydraulic performance.

Due to leakage flow, the flow rate through the impeller is higher than that through the pump. As trimming of the impeller results in a lower head, the leakage flow will be reduced after trimming. The influence of the difference in leakage flows between original and trimmed impellers on the hydraulic characteristics has been studied in Ref. [21], where an extension of the difference method that accounts for the leakage flow rate is formulated. In this extension, the dependence of the leakage flow rate on the impeller head has been described, employing relations given in Refs. [23] and [24]. Models have been developed for the leakage path in the multi-stage pump for which the trim was large (12.23%). For the case with the large trim, this extended model yielded a slightly more accurate prediction for the effect of trimming, but very detailed information on leakage paths and clearances is required that generally may not be available.

5 Conclusions

Prediction methods from literature, together with a new prediction method, have been used to quantify the effect of trimming on the hydraulic performance of low specific-speed centrifugal pumps. The predictions of these methods have been compared to measured effects of trimming for an extensive set of pumps, with different trimming percentages and specific speeds.

The developed difference method is more detailed than the other methods (that are based on the outlet diameter, and on the inlet diameter for the inlet-to-outlet scaling method), as it takes into account losses $\Delta H(Q)$ (that represent the difference between the theoretical head curve and the measured head curve with the untrimmed impeller) as well as more detailed geometrical information (number of blades Z, axial width b_2, blade angle β_2, and blade thickness t_2 at the trailing edge) that is often available as as information from computer-aided design-systems or as drawings. No new empirical parameters are required for the difference method.

The methods have been evaluated based on whether or not the method may frequently lead to overtrimming (negative errors) and on the magnitude of the errors in the predictions for flow rates in the range of between 80% to 110% of the BEP.

The constant-width scaling method, Eq. (2), and the constant-area scaling method with correction, Eq. (4), often lead to overtrimming. The geometrical scaling method does not lead too overtrimming, but it is inaccurate, except for small to moderate trimming percentages (as also noted in Ref. [1]). In comparison to the inlet-to-outlet scaling method, Eq. (6), the difference method generally does not lead to overtrimming (fewer cases with negative errors) and is more accurate. Hence, the difference method proposed here constitutes a balanced improvement in comparison with the methods from literature.

A prediction method for the effect of trimming that is based on time-consuming CFD-based simulations may not be more accurate than simpler methods, such as the difference method proposed here, for low specific-speed pumps.

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