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ERP Positive Displacement Pumps - Experimental Validation of a Type-Independent Efficiency Model

PIF- positive displacement pumps

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ABSTRACT

The European Union pursues ambitious objectives reducing greenhouse gas emissions. Thus, energy related products (ERP) are regulated by the European Union's Ecodesign Directive in terms of energy efficiency. Until now positive displacement pumps are not involved. However, considering the EU strategy in similar domains, the probability is high that the Ecodesign Directive will focus on positive displacement pumps in the future due to the large quantity of sold pumps and the collective energy consumption.

Positive displacement pumps are characterized by their wide range of applications, e.g. mobile hydraulics, chemical, or food industry, with very different application-relevant requirements (low pulsation, high precision etc.). For this reason, a variety of pump designs exist, e.g. piston pumps, gear pump or screw pumps.

From a scientific and technical point of view, legal requirements are considered adequate if they take into account the physical behavior of the machines as well as the application-relevant requirements. Against this background, the authors presented a physically based approach towards an application-related efficiency guideline at IREC 2016 considering a methodology to identify the energetically relevant applications of positive displacement pumps and introducing a physically based, type independent and easy to apply efficiency model.

Since 2017, within the framework of an AiF project, the authors collaborate with the VDMA to validate the above mentioned efficiency model by means of precise experimental measurements. Until the submission of this paper, efficiency measurements are conducted on gear pumps, screw pumps and rotary piston pumps. The full paper presents the validation results and discusses the conclusions in the context of a future energy efficiency guideline of positive displacement pumps.

1 INTRODUCTION

The European Union pursues ambitious objectives reducing greenhouse gas emissions. This applies not only to the energy production sector, e.g. the transition to renewables energies, but also to the energy consumption in the EU. Thus, energy related products (ERP) are regulated by the European Union's Ecodesign Directive in terms of their energy efficiency since 2009 [1]. Until now, more than 40 product groups have already been considered. However, positive displacement pumps are not involved in the Ecodesign Derective, yet, and will not be considered in the near future according to the actual working plan 2016-2019. Taking the EU strategy in similar domains into account, there is still a possibility that the Ecodesign Directive will focus on positive displacement pumps in the medium-term future. On the one hand, there is a large quantity of sold positive displacement pumps and, hence, the collective energy consumption is probably not negligible. On the other hand, in view of the climate change and its negative consequences, the European Union's efforts to reduce green house gas emissions are likely to increase. From this point of view, the issue remains relevant and of high priority for pump manufacturers.

Against this background, the Verein Deutscher Maschinen und Anlagenbauer e.V. (VDMA) and the Chair of Fluid Systems collaborate since 2014 researching proactively into the design of a future energy efficiency guideline for positive displacement pumps. Two joint projects investigated a uniform efficiency description of different types of positive displacement pumps [2] and a methodology to determine energy-relevant applications of positive displacement pumps. The successful application of a new type-independent efficiency model on manufacturer's data and a concept of data acquisition and analysis identifying the energetically relevant applications of positive displacement pumps were presented on the International Rotating Equipment Conference (IREC) 2016 in Düsseldorf, Germany [3]. The conclusion from a scientific and technical point of view is that adequate legal requirements need to take the physical behavior of positive displacement pumps and application-relevant requirements into account.

Two open research questions of the two joint projects were, firstly, the experimental validation of the type-independent efficiency model and, secondly, the investigation on the influence of manufacturing uncertainty on the efficiency of positive displacement pumps. Since 2017 VDMA and Chair of Fluid Systems have been pursing these two research questions as part of a project of Arbeitsgemeinschaft industrieller Forschung (AiF). This project contains two major investigations: firstly, the Chair of Fluid Systems carries out precise efficiency measurements on different types of positive displacement pumps, i.e. gear pumps, spindle screw pumps, rotary lobe pumps, progressive cavity pumps and piston diaphragm pumps, of varying size and at various operating conditions. Secondly, pump manufacturers carry out efficiency measurements on pumps of identical size investigating the influence of manufacturing uncertainty on the efficiency. Until the submission of this paper, the measurements at the Chair of Fluid Systems contain 40 pumps of four different pump types and more than 10000 operating points.

This paper presents the results of the above mentioned AiF project and is structured as follows: The paper begins with a brief description of the type-independent efficiency model of positive displacement pumps. In the third section, the experimental setup comprising two test rigs is described in detail. Subsequently, the results of the efficiency model validation on gear pumps, spindle screw pumps and rotary lobe pumps are presented and discussed. The fifth section presents an approach for the description of manufacturing uncertainty based on the type-independent efficiency model. The paper closes with conclusions and an outlook on future work.

2 TYPE-INDEPENDENT MODEL OF POSITIVE DISPLACEMENT PUMPS

In earlier publications [2,3,4,5] the authors have already derived and discussed the type-independent efficiency model. For reasons of traceability of the validation results, the efficiency model is explained again in detail in this section. Therefore, this section is identical to earlier publications.

Of major importance is the isentropic efficiency η that represents a measure of the energetic quality of a pump. Based on the first law of thermodynamics, for a time averaged stationary and thermally isolated machine the isentropic efficiency is defined as the hydraulic power divided by the shaft power P_S. The hydraulic power is obtained as the product of the volume flow Q_1 at the inlet of the pump, the discharge pressure Δp and a correction factor that depends on the compressibility κ. Considering the shaft power is the product of the shaft torque M_S and the rotational speed n, one obtains the well-known definition of the efficiency η as

$$\eta \coloneqq \frac{Q_1 \Delta p}{2\pi M_S n}\left(1 - \frac{\kappa \Delta p}{2}\right). \tag{1}$$

In the following, the influence of the compressibility is neglected due to low discharge pressures under 100 bar which leads to $\kappa\Delta p \ll 1$. Consequently, the volume flow rate $Q_1 = Q$ is assumed to be constant for $\kappa\Delta p \ll 1$. Extending equation (1) with the displacement volume V, the efficiency can be written as the product of the volumetric efficiency η_{vol} and the mechanical-hydraulic efficiency η_{mh}

$$\eta = \eta_{vol}\eta_{mh}, \quad \eta_{vol} := \frac{Q}{nV}, \quad \eta_{mh} := \frac{\Delta pV}{2\pi M_S}. \qquad (2)$$

Both partial efficiencies can be represented as a function of the respective responsible loss: for the volumetric efficiency, this is the leakage Q_L. Taking the theoretical volume flow rate $Q_{th} = nV = Q + Q_L$ into account, the volumetric efficiency η_{vol} can be written as

$$\eta_{vol} := \frac{Q}{nV} = 1 - \frac{Q_L}{nV}. \qquad (3)$$

The friction torque M_{mh} represents the mechanical-hydraulic losses. Considering the shaft torque is the sum of the hydraulic torque $M_{hyd} = \Delta pV/2\pi$ and the friction torque M_{mh}, gives the mechanical-hydraulic efficiency η_{mh}

$$\eta_{mh} := \frac{\Delta pV}{2\pi M_S} = \frac{1}{1+2\pi\frac{M_{mh}}{\Delta pV}}. \qquad (4)$$

Following the above approach, a description of the losses, i.e. the leakage Q_L and the friction torque M_{mh}, results in a description of the volumetric and mechanical-hydraulic efficiency as well as total efficiency. On this basis, a dimensional analysis is applied, the foundation of our modeling and of similarity. The procedure is as follows:

Firstly, all major influencing variables on the losses are determined. The following six influencing variables are considered: the operational parameters discharge pressure Δp and rotational speed n, the properties of the pumping medium density ϱ and kinematic viscosity ν, and the geometric parameters displacement volume V and average gap height $\bar{s}$ of the pump. The average gap height $\bar{s}$ is a newly introduced size representing an average height of the several different gaps of positive displacement pumps. It is motivated by the analogy of a hydrodynamic journal bearing with the average height of the lubrication gap $\bar{h}$, i.e. $\bar{s}$ is interpreted as $\bar{h}$. Hence, $\bar{s}$ is constant for one single pump und independent of the operating conditions, as long as wear does not occur. The characteristic length of the pump is $V^{1/3}$.

Secondly, performing a dimensional analysis reduces the number of model variables and, thus, simplifies the model while maintaining the physical significance [6]. This yields five dimensionless variables characterizing the operating state of a pump: The specific pressure Δp^+, Reynolds number Re, and relative gap size ψ are the independent dimensionless variables and are defined as

$$\Delta p^+ := \frac{\Delta p}{\nu^2 \varrho V^{-2/3}}, \ Re := \frac{nV^{2/3}}{\nu}, \ \psi := \frac{\bar{s}}{V^{1/3}}. \qquad (5)$$

The relative gap size ψ is constant for one single pump similar to the average gap height $\bar{s}$. Furthermore, both the leakage Q_L and the friction torque M_{mh} are also represented by dimensionless variables, the specific leakage Q_L^+ and the specific friction torque M_{mh}^+. These are the dependent dimensionless variables and are the defined as

$$Q_L^+ := \frac{Q_L}{\nu V^{1/3}}, \ M_{mh}^+ := \frac{M_{mh}}{\Delta pV}. \qquad (6)$$

Thus, the specific leakage $Q_L^+ = Q_L^+(\Delta p^+, Re, \psi)$ and the specific friction torque $M_{mh}^+(\Delta p^+, Re, \psi)$ are functions of the specific pressure, Reynolds number and relative gap. The mathematical description of these functions need to be determined, which leads directly to the descriptions of the volumetric, of the mechanical-hydraulic, and of the total efficiency:

$$\begin{aligned} \eta_{vol} &= 1 - \frac{1}{Re} Q_L^+(\Delta p^+, \psi), \\ \eta_{mh} &= \frac{1}{1+\frac{2\pi}{1-\kappa\Delta p/2} M_{mh}^+(\Delta p^+, Re, \psi)}, \\ \eta &= \frac{1-\frac{1}{Re}Q_L^+(\Delta p^+, \psi)}{1+\frac{2\pi}{1-\kappa\Delta p/2} M_{mh}^+(\Delta p^+, Re, \psi)}. \end{aligned} \qquad (7)$$

In [4] the authors illustrate that the specific leakage $Q_L^+(\Delta p^+, Re, \psi)$ can be approximated by the semi empirical model.

$$Q_L^+ = L_{\Delta p^+} * (\Delta p^+ \psi^3)^m + L_{Re} Re. \qquad (8)$$

The dimensionless coefficients $L_{\Delta p^+}$, L_{Re} and the exponent m are the empiric model parameters. In equation (8) the first term represents a pressure driven flow and the second term represents a drag flow of the leakage.

On the other hand, Schlösser and Hilbrands [7,8,9] introduced a physically based approach for the estimation of the friction torque M_{mh} that represents a linear combination of a pressure-related loss, a viscous friction-related loss and inertia-related loss:

$$M_{mh} = C\Delta pV + R_{\mu}\frac{\mu nV}{\bar{s}} + R_{\varrho}\varrho n^2 V^{5/3}. \qquad (9)$$

Applying the dimensionless variables given by the equations (5) and (6) yields a description of the specific friction torque

$$M_{mh}^{+}(\Delta p^{+}, Re, \psi) = C + R_{\mu}\frac{Re}{\Delta p^{+}\psi} + R_{\varrho}\frac{Re^2}{\Delta p^{+}}. \qquad (10)$$

C, R_{μ} and R_{ϱ} are dimensionless loss coefficients of the different loss terms.

Altogether, there are six empiric model parameters, namely $L_{\Delta p^+}$, L_{Re}, m, C, R_{μ} and R_{ϱ}, that need to be identified, i. e. calibrated, using measurement data. The parameter identification is based on robust non linear regression. In summary, the semi-analytical equations (8) and (10) give the mathematical formulas which are applied on the positive displacement pumps in this paper.

3 EXPERIMENTAL SETUP

The experimental investigations were carried out on two separate experimental test rigs with a nominal shaft power of 75 kW (test rig A) for screw pumps and 22 kW (test rig B) for gear pumps and rotary lobe pumps. Both test rigs and the methods of testing are in accordance with ISO 4409 [10]. Furthermore, test rig B was described for the first time by the authors in [4].

Pressure and temperature are measured at the inlet and outlet of the pump. A piezo resistive sensor and a Pt-100 resistance thermometer are used, respectively. A torque meter with built-in speed sensor operating on the strain gage principle measures the shaft torque and rotational speed of the pump. The volume flow rate at the outlet of the pump is measured by a screw type flow meter. On test rig A the pressure variation is induced by means of a proportional pressure relief valve. On test rig B an electric ball valve in combination with a needle valve is used. Figure 1, 2 and 3 show the principal hydraulic circuit diagram of both test rigs and the two test rigs for the efficiency measurements.

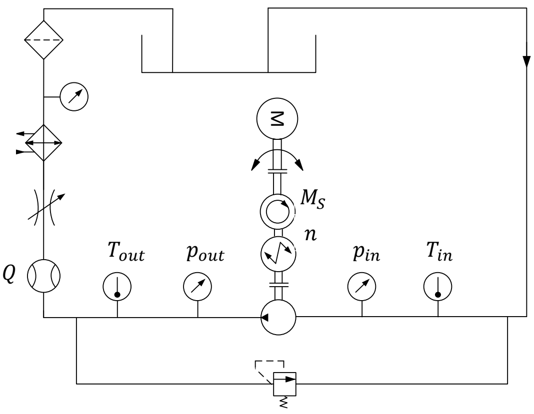

Figure 1. Hydraulic circuit diagram of the test rigs.

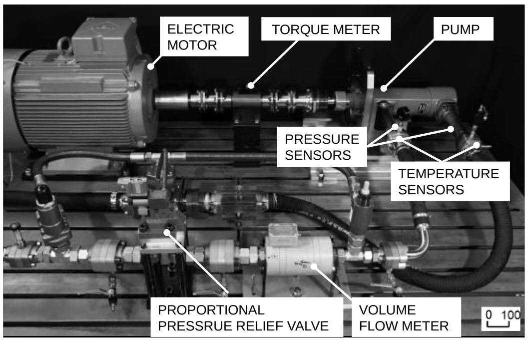

Figure 2. Test rig A with a nominal power of 75 kW.

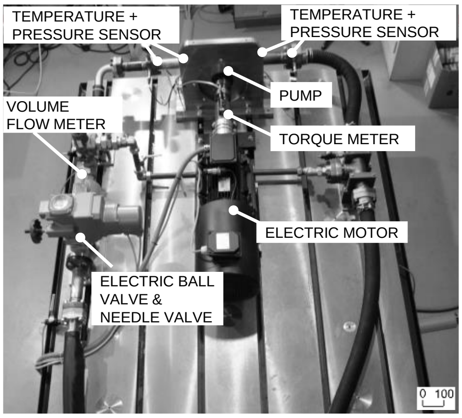

Figure 3. Test rig B with a nominal power of 22 kW.

The measurement equipment used in the investigations predominantly meets the accuracy requirements of class A (see ISO 4409 [10]). Table 1 gives a summary of the accuracy of the used measurement equipment.

Table 1. Measurement ranges and uncertainty of measured operation variables (MW= measured value, FS=full scale).

MEASURED VARIABLES	TEST RIG A / B MEASUREMENT RANGE	TEST RIG A/B MEASUREMENT ACCURACY
p_{in}	$0 \ldots 2.5$ bar	0.15 % FS
p_{out}	$0 \ldots 160$ / 25 bar	0.15 % FS
Q	400 l/min	0.5 % MW
M_S	200 / 50 Nm	0.1 % FS
n	3600 rpm	0.1 % MW
T_{in}, T_{out}	$0 \ldots 100$ °C	± 0.15 °C

The pump fluids used in the investigations are four different hydraulic oils: Shell Tellus 22, 46, 100 and Shell Morlina 10 at a temperature of 40°C. The viscosity-temperature curves and the density-temperature curves of all four hydraulic oils were measured with a highly accurate glass capillary viscometer and a density meter.

4 VALIDATION RESULTS

The validation of the type-independent efficiency model includes the comparison of measurements with model predictions for a spindle screw pump, a gear pump and a rotary lobe pump. On the basis of the measured pump characteristics including two hydraulic oils, the six model parameters $L_{\Delta p+}$, L_{Re}, m, C, R_μ and R_ϱ are identified. Subsequently, the calibrated models, i.e. equations (8) and (10), are applied to predict each pump characteristic for a third hydraulic oil and a different viscosity. The results presented the two following subsections represent the validation on more than 40 pumps of different size within the above mentioned AiF project. For reasons of confidentiality, the results are presented in a dimensionless representation based on the volumetric inefficiency ε_{vol} and mechanical-hydraulic inefficiency ε_{mh} versus the relative discharge pressure Δp_{rel}. Furthermore, no manufacturer names are mentioned. The definitions of the dimensionless numbers are

$$\varepsilon_{vol} := \frac{Q_L}{nV} = \frac{Q_L^+}{Re},$$

$$\varepsilon_{mh} := \frac{M_{mh}}{M_S} = \frac{2\pi M_{mh}^+}{1+2\pi M_{mh}^+}, \qquad (11)$$

$$\Delta p_{rel} := \frac{\Delta p}{\Delta p_{max}}.$$

Δp_{max} is the maximum operational pressure of each pump.

4.1 Volumetric inefficiency

Figure 4, 5 and 6 present the prediction of the volumetric inefficiency together with the measurements at three different Reynolds numbers for the three different pump types. For all pump types, the prediction of the volumetric inefficiency matches the measurements with high accuracy. The results prove the practicability of the leakage model (cf. equation (8)) in calculating the pump characteristics at different viscosities and, from the manufacturer's point of view, provide a solution for a lack of measurement data.

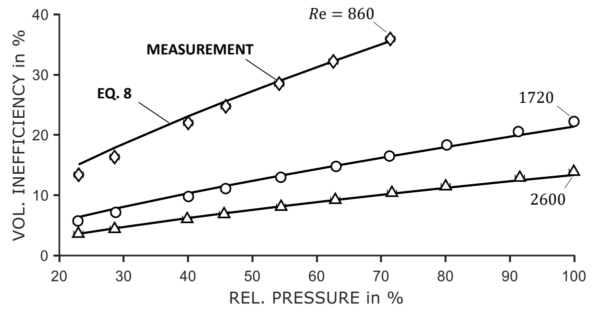

Figure 4. Volumetric inefficiency of a spindle screw pump.

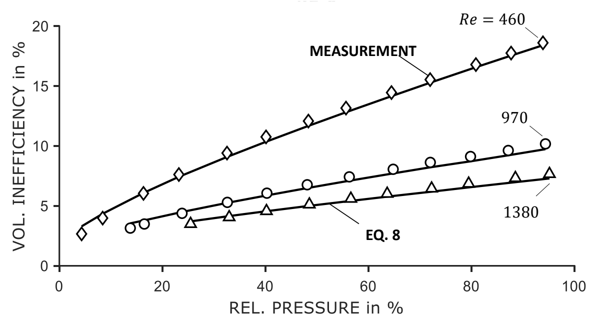

Figure 5. Volumetric inefficiency of a gear pump.

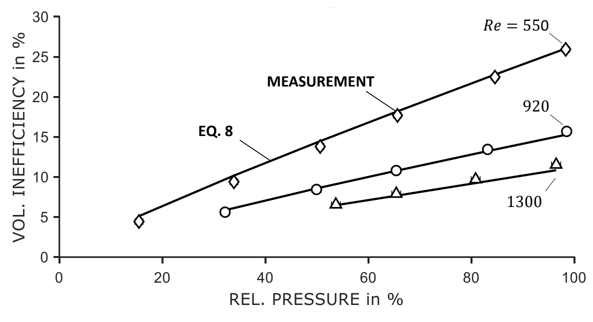

Figure 6. Volumetric inefficiency of a rotary lobe pump.

4.2 Mechanical-hydraulic inefficiency

Figure 7, 8 and 9 show the prediction of the mechanical-hydraulic inefficiency together with the measurements at three Reynolds numbers for the same three pump types. Considering the spindle screw pump and gear pump the prediction of the mechanical-hydraulic inefficiency matches the measurements with good accuracy. Only for low Reynolds numbers and low relative pressures, the deviations from the measurements increase.

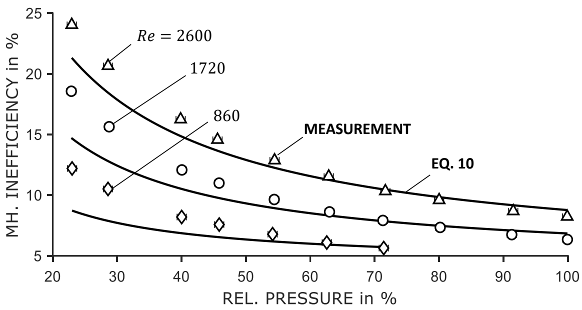

Figure 7. Mechanical-hydraulic inefficiency of a spindle screw pump.

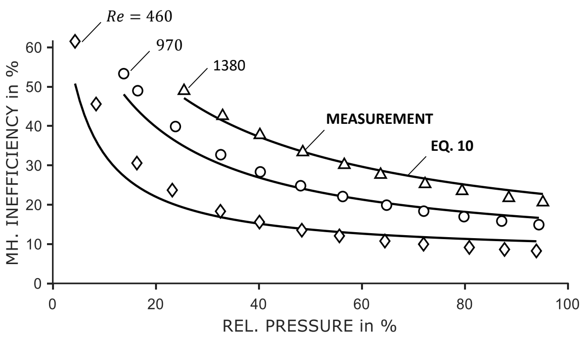

Figure 8. Mechanical-hydraulic inefficiency of a gear pump.

During measurements on the rotary lobe pump, a constant friction torque occurred due to its belt-driven rotors. For this reason, an additional constant term M_c is added in equation (9) and (10). Similar to the six existing model parameters, non linear regression determines M_c.

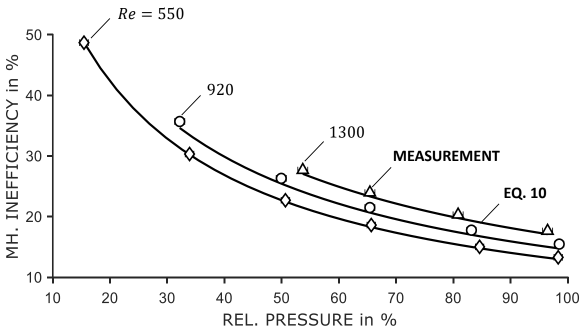

Figure 9. Mechanical-hydraulic inefficiency of a rotary lobe pump.

The prediction of rotary lobe pump's mechanical-hydraulic inefficiency shows the highest accuracy and matches the measurements in all operating points. As with the leakage model, the results also prove the practicability of the friction torque model (cf. equation (10)).

5 REPRESENTATION OF MANUFACTURING UNCERTAINTY

Positive displacement pumps are technically mature machines with high manufacturing effort realising their narrow gaps. Hence, manufacturing uncertainty plays an important role for their manufacturing costs but also for their energy efficiency in operation. Against the background of a future energy efficiency directive for positive displacement pumps, the investigation of manufacturing uncertainty influencing the efficiency is highly relevant. In this section, an approach for representing manufacturing uncertainty on the basis of the type-independent efficiency model is presented. In the context of artificially introduced manufacturing uncertainty the authors already applied this approach on spindle screw pumps [5]. The validation of the approach with other types of positive displacement pumps, i.e. gear pumps, progressive cavity pumps and piston diaphragm, is also part of the above mentioned AiF project, but has not been carried out until the submission of this paper.

The type-independent efficiency model is based on the concept of similarity. However, full geometric similarity in identical positive displacement pumps is never present. The average gap height of a series of identical pumps will always vary from the nominal value due to manufacturing (including assembly) uncertainty. This leads to a scattering of the dimensional and dimensionless characteristic curve, illustrated in figure 10. When correlating the scattering with manufacturing uncertainty it is advantageous to describe the scattering by the relative gap based on the measured pump characteristics instead of detailed geometrical models of the several different gaps in a positive displacement pump. In this way, the relative gap becomes the single quantity to describe the incomplete geometric similarity and the accumulated manufacturing uncertainty that influence the gap heights in a pump. Following this idea, measurements of identical pumps are necessary forming the scattering of the characteristic curve. On such a database, the dimensionless representation of with the specific leakage and specific pressure (see eqn. 5, 6 and 8) needs to be applied and linear regression determines the model parameters $L_{\Delta p^+}$ and m yielding a description of the dimensionless average characteristic curve, cf. figure 10. For reasons of simplicity, in the following, only constant rotating speeds are considered and, thus, the second term of equation (8) representing the drag flow is neglected. Furthermore, the relative gap is considered constant and defined as $\psi_{ref} := 1$ for the average characteristic curve.

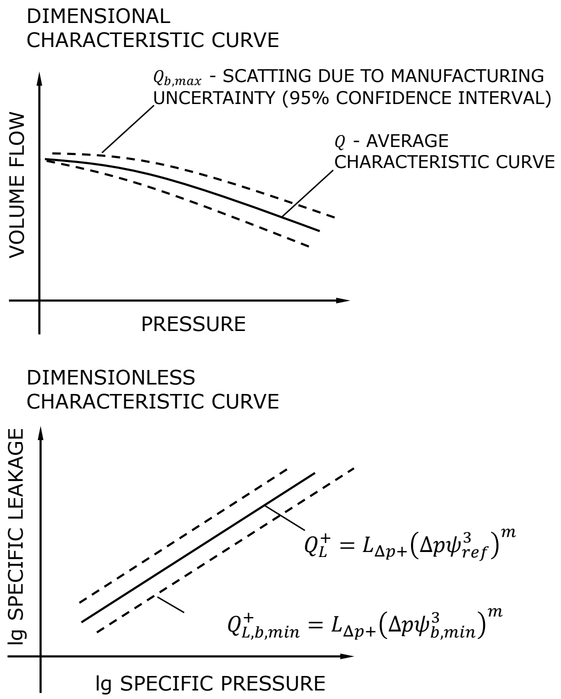

Figure 10. Scattering of the dimensional and dimensionless characteristic curve due to manufacturing uncertainty.

The 95% confidence interval of the dimensionless characteristic curve $z_{0.95,Q_L^+}$ defines the boundaries of the scattering and predominantly depends on the 95% confidence interval $z_{0.95,L_{\Delta p+}}$ of the model parameter $L_{\Delta p^+}$ which is

$$z_{0.95,L_{\Delta p+}} = 1.96\sigma_{L_{\Delta p+}}\,. \tag{12}$$

$\sigma_{L_{\Delta p+}}$ is the standard deviation of $L_{\Delta p^+}$ given by linear regression. Using Gaussian error propagation, equation (8) and (12) give the 95% confidence interval $z_{0.95,Q_L^+}$

$$z_{0.95,Q_L^+} = \pm 1.96\sigma_{L_{\Delta p+}}\left(\Delta p^+ \psi_{ref}^3\right)^m. \tag{13}$$

Aiming to represent $z_{0.95,Q_L^+}$ based on the relative gap a relative consideration is applied, firstly introduced by Pelz et. al. [2] comparing different types of positive displacement pump. For this purpose, the boundary of the 95% confidence interval $z_{0.95,Q_L^+}$ is represented by the relative gap ψ_b (ψ_b can be larger or less than ψ_{ref}) and one obtains

$$z_{0.95,Q_L^+} = \left|\left(\psi_b^{3m} - \psi_{ref}^{3m}\right) L_{\Delta p+} \Delta p^{+m}\right|. \tag{14}$$

Equalising equation 13 and 14 yield

$$\left|\frac{\psi_b^{3m}-\psi_{ref}^{3m}}{\psi_{ref}^{3m}}\right| = \frac{1.96\sigma_{L_{\Delta p+}}}{L_{\Delta p+}} \tag{15}$$

which can be solved after ψ_b with $\psi_{ref} := 1$ and one obtains

$$\psi_b = \left(\pm\frac{1.96\sigma_{L_{\Delta p+}}}{L_{\Delta p+}} + 1\right)^{1/(3m)}. \tag{16}$$

Equation (16) gives the calculation formula for ψ_b representing the relative gap limiting the 95% confidence interval of the dimensionless characteristic curve $z_{0.95,Q_L^+}$. Applying equation (16) on the dimensional characteristic curve (cf. fig. 10) with $\psi_{ref} := 1$ the characteristic curve Q and the boundary Q_b of the scattering of the characteristic curve yield

$$Q = nV - \nu V^{1/3} L_{\Delta p+}\left(\frac{\Delta p V^{2/3}}{\nu^2 \varrho}\right)^m - L_{Re} nV \tag{17}$$

and

$$\begin{aligned} Q_b &= nV \\ &\quad - \nu V^{1/3} L_{\Delta p+}\left(\frac{\Delta p V^{2/3}}{\nu^2 \varrho}\right)^m - L_{Re} nV \\ &\quad \pm 1.96\sigma_{L_{\Delta p+}} \nu V^{1/3}\left(\frac{\Delta p V^{2/3}}{\nu^2 \varrho}\right)^m. \end{aligned} \tag{18}$$

Following this argumentation, the relative gap is a suitable characteristic measure of manufacturing uncertainty. Applied on a broad database of efficiency measurements the presented approach provides a basis for describing manufacturing uncertainty in a future energy efficiency directive. Furthermore, the approach can be of high value for the monitoring process of manufacturers rating newly manufactured pumps against the average characteristic curve. Measuring the whole characteristic curve of newly manufactured pumps on a random basis, the manufacturing process can be monitored reasonably by the relative gap ψ of each random sample.

6 CONCLUSIONS

The validation results prove that the presented type-independent efficiency model is able to represent a positive displacement pump's characteristic over a wide operating range. The model predictions of the volumetric and mechanical-hydraulic inefficiency show high accuracy for all considered pump types and are of high value for the pump manufacturers. Furthermore, this paper presents an approach to describe manufacturing uncertainty based on only one single quantity: the relative gap. In summary, the type-independent efficiency model gives a validated basis for an efficiency description in a future efficiency directive for positive displacement pumps.

However, it is necessary not only to keep the focus on the pump but also to consider an efficiency directive in a broader context. At this point, water pumps serve as a useful example. In the latest EU ecodesign regulation for water pumps not only the single pump but the extended product of pump, electric motor and frequency converter are considered and rated based on a specific load profile. This gives a direction of proactive research into positive displacement pumps in the future.

7 NOMENCLATURE

C	pressure-related loss coefficient
$\bar{h}$	average height of lubrication gap
$L_{\Delta p+}$	dimensionless leakage coefficient
L_{Re}	dimensionless leakage coefficient
m	exponent for power law of specific leakage
M_{hyd}	hydraulic torque
M_{mh}	friction torque
M_{mh}^{+}	specific friction torque
M_S	shaft torque
n	rotational speed
Δp	discharge pressure
Δp^{+}	specific pressure
Δp_{max}	maximum discharge pressure
Δp_{rel}	relative discharge pressure
p_{in}	pressure at inlet of pump
p_{out}	pressure at outlet of pump
P_S	shaft power
Q	volume flow rate
Q_1	volume flow rate at inlet of pump
Q_b	boundary of scattering of the characteristic curve
Q_L	leakage
Q_L^{+}	specific leakage
Q_{th}	theoretical volume flow rate
R_{μ}	viscous friction-related loss coefficient
R_{ϱ}	pressure-related loss coefficient
Re	Reynolds number
$\bar{s}$	average gap height
V	displacement volume
$z_{0.95,L_{\Delta p+}}$	95% confidence interval of the model parameter $L_{\Delta p+}$
$z_{0.95,Q_L^{+}}$	95% confidence interval of the dimensionless characteristic curve
ε_{mh}	mechanical-hydraulic inefficiency
ε_{vol}	volumetric inefficiency
η	total efficiency
η_{mh}	mechanical-hydraulic efficiency
η_{vol}	volumetric efficiency
κ	compressibility
μ	dynamic viscosity
ν	kinematic viscosity
ϱ	density
$\sigma_{L_{\Delta p+}}$	standard deviation of the model parameter $L_{\Delta p+}$
ψ	relative gap
ψ_b	relative gap of boundary of 95% confidence interval
ψ_{ref}	reference relative gap

8 ACKNOWLEDGEMENTS

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