An industrial method for performance map evaluation for a wide range of centrifugal pumps

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Abstract

Centrifugal pumps are designed and manufactured in order to be fitted to installations and work over a wide range of operating conditions. In such cases the prediction of performance constitutes an important challenge for the pump designer. The challenge becomes particularly difficult when it is necessary to predict the performance of different types of centrifugal pumps varying from low to high volume flow rates. Even if one possesses the rig to measure the performance of a pump, it is useful and time saving to predict numerically the overall pump characteristics. The present method is a simple and easy to apply numerical tool for pump performance curve estimation. It requires a minimum of pump geometrical data and it can be advantageous to pump designers providing them with an initial performance curve estimation during the design process, before they advance to the detailed design of the pump and its experimental verification on the test rig. From the cases examined, it is concluded that the proposed method provides a satisfactory approximation of industrial centrifugal pumps' performance curves, constituting a potential tool for pump manufacturers.
Keywords: performance curve, characteristic line, centrifugal pump, numerical prediction.

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1 Introduction

Prediction of centrifugal pump performance constitutes an important challenge when a pump has to be manufactured in order to be fitted in a given installation and to work over a wide range of operating conditions, Samani [1]. The challenge becomes particularly difficult when is needed to predict the performance of different types of centrifugal pumps varying from low to high volume flow rates Pfeiderer [2]. Characteristic curves are not always available to evaluate the adequacy of the pump's performance for a particular situation, Engeda [3]. Even if one possesses the rig to measure the performance of a pump, it is useful and time saving to predict numerically the overall pump characteristics.

Significant numerical work was done over the past years to estimate the flow and the performance characteristics of centrifugal pumps [1–10]. An interesting method was presented by Engeda [3] for the Head prediction. It has been demonstrated by Engeda [3] that predictions based on the Euler's method and airfoil theory, sometimes produce unrealistic results. Sophisticated three-dimensional methods including the interaction between impeller and volute, Lakshminarayana [11] are time consuming, require detailed three-dimensional geometrical data of the impeller and volute and they are not suited as an engineering tool for performance prediction, but for the detailed flow analysis inside the pump.

The present study presents a fast method requiring only a few pump geometrical data to estimate performance characteristics of centrifugal pumps. Not only the Head, but also the overall efficiency and the required power of the motor to drive the pump are estimated. Numerical predictions are compared to experimental data that was either obtained in the test rig or found in the literature, for centrifugal pumps delivering low, medium and high volume flows. The results show that the proposed method can be used as a tool to provide a quick assessment of performance curves to the pump designer.

2 Numerical method

The maximum head produced by a centrifugal pump corresponds to throttling conditions, where the volume flow is zero, Bohl [14]. The maximum theoretical head is proportional to the square of the impeller rotational speed and of the impeller tip diameter, i.e. available head can be approximated as:

$$H_{theor} \approx \left(n \cdot D_2\right)^2 \tag{1}$$

Taking into account that according to Japikse [5] the main sources of losses inside a centrifugal pump are mechanical losses, impeller losses, disk friction losses and leakage losses in the gap between impeller and casing, the maximum available head can be written as:

$$H_{\max} = K_1 \cdot \left[\left(n \cdot D_2\right)^2\right] \tag{2}$$

where n is the impeller rotational speed in rev/*s and* D_2 is the impeller tip diameter in *m*. The coefficient K_1 involved in this formula is a loss coefficient at throttling. It is defined as $K_1 = 0.6$ that it can be able to capture all types of losses referred previously at throttling conditions (i.e. when $Q = 0$). The constant K_1 has the units so that the Head is expressed in S.I. units. Similar choice of the loss coefficient at throttling conditions was done in the past by Stepanoff [15].

To simulate the pump behaviour for all other operation points up to the maximum flow that the pump can deliver, the non-dimensional flow rate, namely ξ is introduced. The coefficient ξ is semi-empirical and is defined by the volume flow at any operating point as well as easily attainable pump data, such as D2x,, b_2, d_P, n:

$$\xi(Q) = K_2 \cdot \frac{Q}{d_p \cdot b_2 \cdot n \cdot D_{2x}} \tag{3}$$

It should vary from 0 at throttling conditions (where $Q = 0$) to a value close to 1 indicating at that point the maximum estimated flow delivered by the pump. The non-dimensional coefficient K_2 is defined as $K_2 = 0.3$ to approximate the maximum predicted flow rate as close as possible to the one obtained in the test rig.

Due to the fact that a lot of pump manufacturers use the pump discharge diameter d_P and the impeller tip diameter D_{2x} to group pump categories, these two geometrical data were deliberately used in equation (3).

Characteristic curves based on experimental data by Vlachakis [16], Inoue and Cumpsty [17], show a slight curvature in the area of throttling where the volume flow is zero and an almost linear behaviour elsewhere. This behaviour was thought to be captured in the best way by using a cosine function to an exponent. The exponent 0.2 guarantees an almost linear behaviour of the H-Q curve, for the values of ξ of interest, $\xi \in [0,1]$, that is close to the real (measured) performance of centrifugal pumps:

$$H = K_1 \cdot \left[\left(n \cdot D_{2X} \right)^2 \right] \cdot \left[\cos\left(\frac{\pi}{2} \cdot \xi \right) \right]^{0,2} \tag{4}$$

Additionally, it can be easily verified that the slope $dH/d\xi$ is always negative, for the values of $\xi \in [0,1]$ which means that the proposed approximation warrantees monotonic behaviour of the performance curve.

3 Validation of the method

The numerical model described in the section 2 was in a first stage validated against experimental data found in the literature. As test cases, the same pump data used by Amminger and Bernbaum [8] and by Yedidiah [18] were chosen to be compared to the numerical results obtained by the present work.

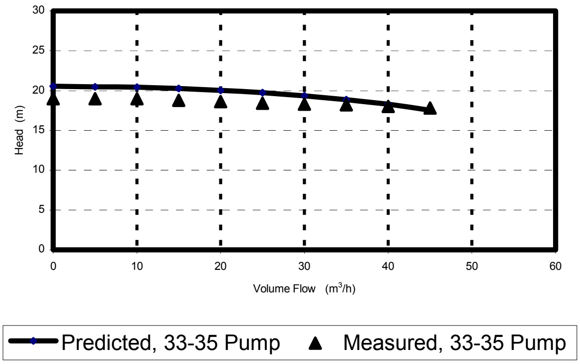

Figure 1: Predicted versus experimental performance curves for the centrifugal pump named 33-35 in Amminger and Bernbaum [8].

Figure 1 shows the comparison between predictions and experimental data from the so-called 33-35 pump for which experimental data presented by Amminger and Bernbaum [8]. It can be seen from this figure that Head predictions show very good agreement to experimental data. In the same article there are experimental data of the so-called 26-14 pump. Figure 2 shows the comparison between numerical and experimental data. The agreement is good for the maximum head as well as the head distribution up to maximum capacities.

Figure 3 presents the comparisons between predictions and experimental data for two different tip impeller diameters, for D_{2x}=136 *mm* and for D_{2x}=16m *mm*. Comparing numerical and experimental data, one observes a good agreement for all the range of flow capacities.

4 Comparison of predictions to measurements obtained in the test rig

After it had been validated, the present numerical model was applied to more than 30 different pumps tested in the test rig of the University of Chalkis. Since it is not possible to present all these results here, only some typical cases were selected that reveal the applicability limits as well as the constraints of the present method. The experimental set-up will not be commented in the present work because the purpose here is the presentation of the prediction method and not of the measurement chain. The results obtained are grouped in three sections; pumps that deliver low, medium and high volume flows.

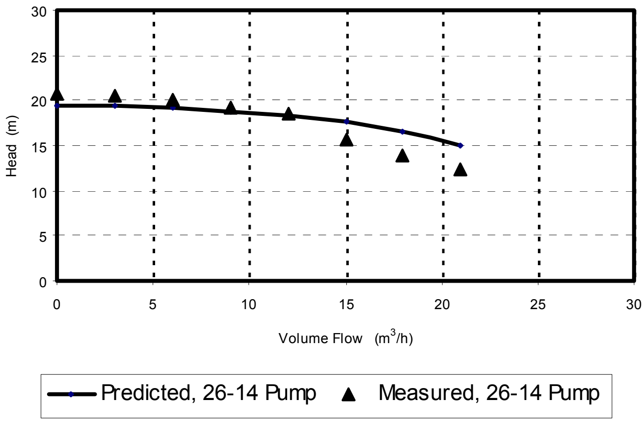

Figure 2: Predicted versus experimental performance curves for the centrifugal pump named 26-14 in Amminger and Bernbaum [8].

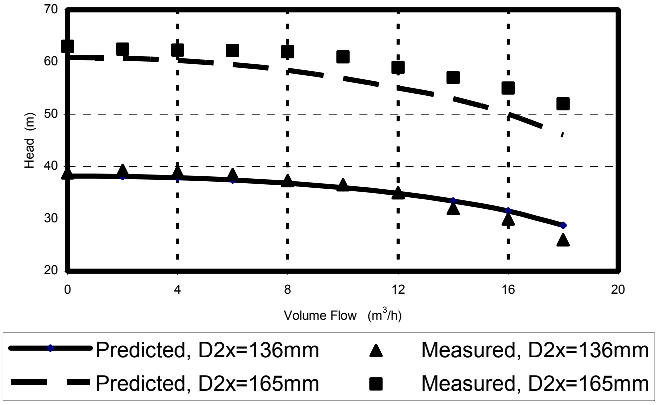

Figure 3: Predicted versus experimental performance curves for the centrifugal pump of figure 2 of Yedidiah [18].

4.1 Low volume flow pumps

A centrifugal pump category with nominal impeller tip diameter D_2= 200 *mm* having discharge diameter d_p= 32 *mm,* running at 1450 rpm. Two sets of centrifugal impellers were used: One having D_{2x}= 215 *mm* and another having

D_{2x}= 185 *mm*. Numerical predictions show an over- prediction of the Head of the pump for both diameters used, when compared to experimental data, figure 4.

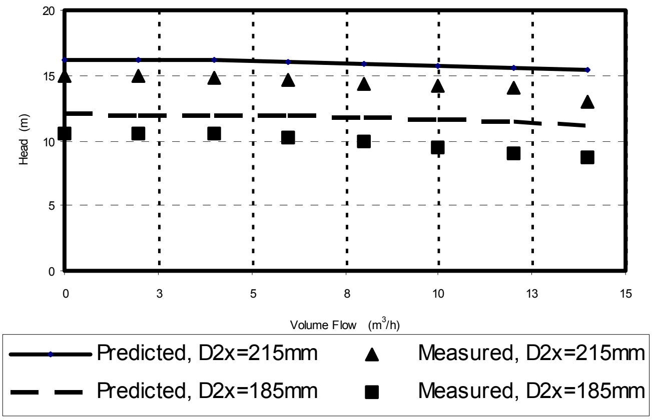

Figure 4: Predicted versus experimental performance curves for centrifugal pumps with nominal D_2=200 *mm*, d_p=32 *mm* running at 1450 *rpm*.

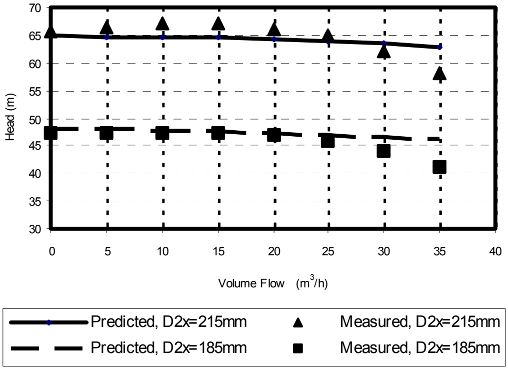

Figure 5: Predicted versus experimental performance curves for centrifugal pumps with nominal D_2=200 *mm*, d_p=40 *mm* running at 2900 *rpm*.

4.2 Medium volume flow pump

Comparisons between predictions and experimental data for another medium flow pump category are shown in figure 5. This centrifugal pump has nominal impeller tip diameter D_2=200 *mm*, discharge diameter d_p=40 *mm* running at 2900 *rpm.* Two sets of centrifugal impellers were tested: One having D_{2x}=215 *mm* and another having D_{2x}=185 *mm*. The Head prediction is in good agreement to experimental data.

4.3 High volume flow pump

Figure 6 shows the predicted versus the experimental head for a centrifugal pump with nominal impeller tip diameter D_2= 400 *mm*, discharge diameter d_p=250 *mm,* running at 1480 *rpm*. Two cases were examined: one with D_{2x}=400 *mm* and another one with D_{2x}= 350 *mm*. The comparison between the predicted and measured Head is again very good for all the range of volume flows.

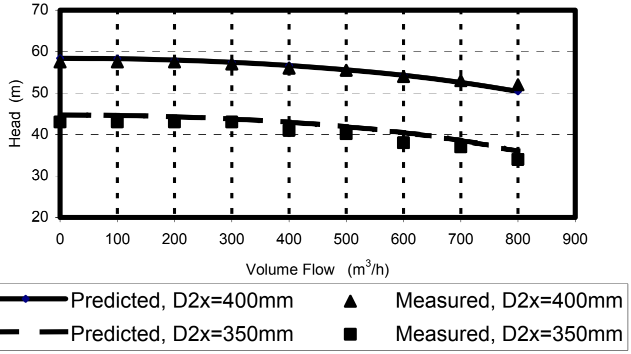

Figure 6: Predicted versus experimental performance curves for centrifugal pumps with nominal D_2=400 *mm*, d_p=250 *mm* running at 1480 *rpm*.

5 Conclusions

A simple and fast method was presented attempting to predict performance curves of industrial centrifugal pumps. For all centrifugal pumps examined, delivering low, medium and high volume flows, the same semi-empirical coefficients and equations were deliberately used in the model. To validate the model, centrifugal pumps for which experimental data were found in the literature, were tested with satisfactory results. Comparisons between numerical and experimental data obtained in the test rig show that the proposed model can satisfactorily predict performance characteristics of centrifugal pumps, for the cases examined. In the most of the cases it seems that the present numerical model over-predicts the Head distribution for low volume flow pumps, while it gives better predictions for medium and high volume flow pumps. This feature of the model seems to underestimate the losses at throttling for low capacity centrifugal pumps, whereas for medium and maximum capacity pumps it provides a better estimation of the maximum head. For the cases examined the assessment of the Head at throttling conditions where Q=0 proved satisfactory using equation (2) whereas the shape and the rate of decrease of the Head as the volume flow is increasing is adequately predicted using equation (4).

The present method is a simple and easy to apply numerical tool for pump performance curves' estimation. It requires a minimum of pump geometrical data and it can be advantageous to pump designers providing them with an initial performance curve estimation during the design process, before they advance to the detailed design of the pump and its experimental verification on the test rig. Furthermore, in cases where pump's characteristics are not available, the present method works as a quick assessment tool to give an educated guess to the question whether a particular pump is suitable to fulfil the installation's requirements.

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