23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

CORRELATIONS FOR THE PREDICTION OF THE HEAD CURVE OF CENTRIFUGAL PUMPS BASED ON EXPERIMENTAL DATA

Natália Argene Lovate Pereira Fleischfresser
Jorge Luiz Biazussi
Antonio Carlos Bannwart
States University of Campinas – UNICAMP – Department of Petroleum Engineering – Campinas - SP
nataliaargene@gmail.com
jorge.biazussi@gmail.com
bannwart@fem.unicamp.br

Abstract. *The hydraulic performance of centrifugal pumps depends on several hydraulic dimensions of the pump, but most of them are not easily accessible. Therefore, the pump's hydraulic performance always has to be informed by the pump manufacturer. However, in order to protect their intellectual property, manufacturers rarely share detailed information about the pump hydraulics with the public. As a consequence, pump users and researchers don't have access to all the data they possibly need. In literature, there are several proposed models that attempt to predict the hydraulic performance of centrifugal pumps. It is also possible to calculate a pump's performance with numerical simulation. However, the accuracy of such models is usually proportional to the number of necessary parameters, those of which may not be easily accessible. In this work, a simple approach available in literature, based on fluid dynamics principles, that predicts a pump's hydraulic performance with only a few accessible hydraulic dimensions, is validated with several experimental data. Eighty tests of different types of pumps, with a large range of specific speeds are considered. From this analysis, correlations among the coefficients of the model equation and the main hydraulic data of the pumps were proposed. Given all the assumptions and simplifications, the objective of this work is to present a method applicable to several pump types that easily provides a prediction of the whole head curve with reasonable error.*
Keywords: *Centrifugal Pumps, Hydraulic Performance, Experimental Validation, Head Curve Prediction*

1. INTRODUCTION

Centrifugal pumps are turbomachines that provide energy increase to the fluid. In order to transport fluid from one place to another, pumps are used to provide a static pressure increase to the flow, which could be useful overcoming a height difference and long distances (pressure losses in the line). Centrifugal pumps are widely used in the industry. Some examples of segments that use centrifugal pumps are Oil & Gas (off-shore, distribution, refineries), power, water and wastewater, pulp and paper and general industry.

During a project conception, it may be necessary to estimate the performance of some equipment in order to assess the project cost and its viability. However, most pump hydraulic dimensions are not easily accessible. As a result, the pump hydraulic performance always has to be informed by the pump manufacturer, who has designed the pump hydraulics. Furthermore, in order to protect their intellectual property, manufacturers rarely share more detailed information about the pump hydraulics with the public. As a consequence, pump users and researchers don't have access to all the data they possibly need.

In literature, there are several proposed models that attempt to predict the hydraulic performance of centrifugal pumps. It is also possible to calculate a pump's performance with numerical simulation. However, the accuracy of such models is usually proportional to the number of necessary parameters – those of which may not be easily accessible, as stated before. Based on this scenario, it would be very useful to predict a pump's hydraulic performance with only a few accessible hydraulic dimensions, so that the dependency on the manufacturer's information would be reduced.

Biazussi (2014) has presented a simple single-phase approach, based on fluid dynamics principles, that defines the head versus flow rate curve. According to this approach, for fluids with low viscosity, the head curve is a quadratic function of the flow rate, as expected. Some assumptions and simplifications were considered, so that the model would only depend on a few accessible hydraulic dimensions and the performance would be predicted with reasonable error.

One of the equation coefficients, k_1, is defined by accessible hydraulic dimensions of the pump. The other coefficients are adjusted by the model in order to reproduce the given tested data. Some tests were held in order to validate this model and also propose correlations among these coefficients and hydraulic dimensions of the pumps.

The results of this work were presented and were considered reliable for these few cases studied. However more experimental data was necessary to fully validate the model.

The current work uses Biazussi (2014)'s work as a starting point. The data of eighty tests was collected so that the aforementioned model could be validated and that its coverage could be defined. Several different types of centrifugal pumps, with a large range of specific speeds, were considered. Furthermore, the mentioned equation coefficients were plotted against the pump specific speed in order to analyze if there was a correlation among them. It was possible to find clear tendencies, even considering all the different types of pump. Considering these tendencies, equations were proposed in order to make it possible to define the head curve against flow based only on the specific speed and main hydraulic dimensions of the pump. The correlation based curves were compared to the tested curves and the difference between them was analyzed.

Given all the assumptions and simplifications, the objective of the current work is to present correlations applicable to several pump types that easily provides a prediction of the pump head curve with reasonable error.

2. LITERATURE REVIEW

According to Gülich (2007), losses arise whenever a fluid flows through (or components move in) a machine. The useful power is therefore always smaller than the power supplied at the pump shaft, where the losses are dissipated into heat. Consequently, the real characteristic curve can be obtained by subtracting the hydraulic losses caused by friction and vortex dissipation from the theoretical head.

The shear stresses created by the velocity gradient in the non-separated boundary layers are responsible for the friction losses. In this work, all of the following were considered as vortex dissipation losses: localized losses, shock losses, losses due to secondary flow and recirculation. Since recirculation causes vortex dissipation, the losses due to this effect will be also considered as hydraulic losses. Non-uniform flow generates losses caused by turbulent dissipation and this is considered the main source of energy loss in pumps. Flow separations and secondary flows also increase the non-uniformity of a velocity distribution and, consequently, the losses.

As already mentioned, in literature, there are several proposed models based on fluid dynamic principles and experimental data that attempt to predict the hydraulic performance of centrifugal pumps. There are also various methods to calculate a pump's performance with numerical simulation. Some of these methods are presented below.

Patel et al. (1981) presents a theoretical model to predict the whole head curve of centrifugal pumps based on fluid dynamic principles, over a wide range of specific speeds, considering various sizes and types of pumps. The head curve is presented as being the Euler head minus friction and shock losses. The results are found to be in good agreement with the experimental data of more than 50 pumps, with an error of ± 3%.

Sun & Tsukamoto (2001) presents a CFD model to predict off-design performance of diffuser pumps, a condition in which the effects of rotor/stator interaction and the pump system characteristics become significant. The predicted head curve is validated by experimental data. The model produces a good prediction of pump off-design performance.

Li et al. (2002) affirms that most published guidelines for the selection and design of centrifugal pumps are based on performance data collected before 1960, on pump designs of the 1920's and 1950's. It analyzes the performance of a modern pump in an attempt to update the information. According to this work, calculating the performance of a modern pump from the available published guidelines may lead to discrepancies of up to 10% from the real performance.

Sun & Prado (2003) presents a single-phase model based on fluid dynamic principles, for different ESP pump types, liquid properties and motor speeds. Friction and shock losses are taken into account. A comparison between the predicted performance and the pump performances based on the Affinity Law is presented.

Asuaje et al. (2005) presents a 3D-CFD simulation of the impeller and volute of a centrifugal pump. This flow simulation is carried out for several impeller vane numbers and the relate volute tongue positions. According to this work, the volute tongue causes an unsymmetrical flow distribution in the impeller, which is confirmed by experimental data. Finally, velocity and pressure fields are calculated for different flow rates.

Cheah et al. (2007) presents a numerical simulation of the internal flow in a centrifugal pump impeller by using a three-dimensional Navier-Stokes code. Both design and off-design conditions are analyzed. The results show that the impeller passage flow at the design point is smooth and follows the curvature of the vane. However, flow separation is observed at the leading edge due to non-tangential inflow conditions. When the centrifugal pump is operating under off-design conditions, unsteady flow develops in the impeller passage. The results are compared to experimental data over a wide range of flow and show good agreement.

Thin et al (2008) presents a method to predict the performance curves of a centrifugal pumps based on fluid dynamic principles. The theoretical head, slip factor, shock losses, recirculation and friction losses are taken into account. The pump considered in this work is a single-stage end suction centrifugal pump, with low specific speed.

Das et al. (2010) presents a performance prediction method for centrifugal submersible slurry pumps, based on fluid dynamic principles. Initially, the head curve for clean water is considered as the theoretical head deducted by the head losses. Then, the effects of solid particle size, specific gravity and concentration on pump slurry flow are shown. Finally, additional head losses due to solid particles in the slurry are predicted and deducted from the clean water head to establish the performance of centrifugal slurry pump. The performance prediction using this method is more accurate around the design flow rate and gives a more accurate prediction of centrifugal slurry pump performance for mostly homogenous slurry than it does for heterogeneous slurry.

Jafarzadeh et al. (2011) presents a general three-dimensional numerical simulation of turbulent fluid flow in order to predict the velocity and pressure fields of a centrifugal pump. A low specific speed, high speed pump is considered.

A commercial CFD code is used. The comparison between the resultant performance curves and experimental data shows acceptable agreement. Furthermore, the effect of the number of impeller vanes on the hydraulic efficiency and the effect of the position of the vanes with respect to the tongue of the volute on the start of the separation are analyzed.

El-Naggar (2013) presents a one-dimensional flow procedure for the prediction of centrifugal pump performance, based on principle theories of turbomachines, such as the Euler equation and the energy equation. The loss at the impeller exit associated to the slip factor and the volute loss are estimated. The predicted curves are consistent with experimental characteristics of centrifugal pumps.

Shah et al. (2013) presents a critical review of the CFD analysis of centrifugal pumps along with the future scopes for further improvements, since in recent years it has been observed a growing availability of computational resources and progress in the accuracy of numerical methods. According to this work, the most active areas of research and development are the analysis of two-phase flow, pumps handling non-Newtonian fluids and fluid-structure interaction. The CFD approach provides many advantages compared to other approaches, but due to its theoretical nature, validation with experimental results is highly recommended.

Biazussi (2014) presents a simple single-phase approach, based on fluid dynamic principles, that defines the head versus flow rate curve. According to this approach, for fluids with low viscosity, the head curve is a quadratic function of the flow rate, as expected. Some assumptions and simplifications were considered, so that the model would only depend on a few accessible hydraulic dimensions and the performance would be predicted with reasonable error.

3. THEORETICAL MODEL

Considering the aim of this work, the theoretical model presented by Biazussi (2014) is chosen as the starting point, due to its simplicity and dependency on few and accessible data of the pumps. This model establishes a method for the interpretation of the head versus flow rate curve of a centrifugal pump. The following premises are considered:

- Homogeneous and single-phase flow
- One-dimensional and permanent flow
- Uniform flow through the impeller
- Constant speed
- Torque due to superficial and field forces are neglected

3.1 Dimensional analysis

The differential pressure is a function of the head provided by the pump, while the head depends on the volumetric flow rate (Q), speed (ω), density (ρ), viscosity (μ), impeller diameter (D_2) and other hydraulic dimensions (L_i).

$$\Delta P = h_1(Q, \omega, \rho, \mu, D_2, L_i) \tag{1}$$

Through dimensional analysis, we can derive the following equations:

Capacity coefficient: $C_Q = \frac{Q}{\omega {D_2}^3}$ (2)

Head coefficient: $C_H = \frac{\Delta P}{\rho \omega^2 {D_2}^2}$ (3)

Viscosity coefficient: $X = \frac{\mu}{\rho \omega {D_2}^2} = \frac{1}{Re_\omega}$ (4)

Geometric coefficients: $\frac{L_i}{D_2}$ (5)

Specific speed: $N_s = \frac{C_Q^{1/2}}{C_H^{3/4}}$ (6)

As a result:

$$C_H = F\left(C_Q, X, \frac{L_i}{D_2}\right) \quad \rightarrow \quad \frac{\Delta P}{\rho \omega^2 {D_2}^2} = H_1\left(\frac{Q}{\omega {D_2}^3}, \frac{\mu}{\rho \omega {D_2}^2}, \frac{L_i}{D_2}\right) \quad \rightarrow \quad C_H = H_1\left(C_Q, X, \frac{L_i}{D_2}\right) \tag{7}$$

The parameter X corresponds to the inverse of the Reynolds number calculated with the rotational speed.

A generic expression for H_1 has to be defined based on physical phenomena. The correlations have to be adjusted to the experimental data. The pump geometry is taken into account by the model adjustment coefficients.

The differential pressure provided by the pump can be calculated by subtracting the losses from the Euler differential pressure:

$$\Delta P = \Delta P_{Euler} - \Delta P_{losses} \tag{8}$$

Where:

$$\Delta P_{Euler} = \rho\omega^2 r_2^2 \left(1 - \frac{Q \cot g \beta_2}{2\pi B_2 \omega r_2^2}\right) \tag{9}$$

And $r_2 = D_2/2$.
The Euler differential pressure can also be written as:

$$\Delta P_{Euler} = \frac{1}{4}\rho\omega^2 {D_2}^2 - k_1 \frac{\rho\omega Q}{D_2} \tag{10}$$

Where $k_1 = \frac{D_2 \cot \beta_2}{2\pi B_2}$ (11)

The hydraulic losses are composed of the friction, localized and distortion losses:

$$\Delta P_{losses} = \Delta P_{friction} + \Delta P_{loc} + \Delta P_{distortion} \tag{12}$$

The friction losses are expressed by an equivalent friction factor, which is composed of a turbulent term and a viscous term:

$$\Delta P_{friction} = f \frac{L_i}{D_2}\frac{\rho}{2}\left(\frac{Q}{A_n}\right)^2 = k_a f \rho\omega^2 {D_2}^2 \left(\frac{Q}{\omega {D_2}^3}\right)^2 \tag{13}$$

$$f = k_2^* \frac{\mu D_2}{\rho Q} + k_3^* \left(\frac{\mu D_2}{\rho Q}\right)^n \tag{14}$$

Where $A_n = 2\pi\bar{r}\bar{B}$, $k_2 = k_a {k_2}^*$ and $k_3 = k_a {k_3}^*$. Therefore, it follows that:

$$\Delta P_{friction} = \left[k_2 \left(\frac{\mu D_2}{\rho Q}\right) + k_3 \left(\frac{\mu D_2}{\rho Q}\right)^n\right] \rho\omega^2 {D_2}^2 \left(\frac{Q}{\omega {D_2}^3}\right)^2 \tag{15}$$

Where n expresses the viscous effect in high flow rate and is always smaller than 1.
According to the dimensional analysis, the friction head losses are written as:

$$C_{H_{friction}} = \left[k_2 \left(\frac{\mu D_2}{\rho Q}\right) + k_3 \left(\frac{\mu D_2}{\rho Q}\right)^n\right] {C_Q}^2 \tag{16}$$

The localized losses are considered purely inertial and independent of the viscosity. They are represented by the sum of the dissipative losses that occur in the inlet and outlet of the impeller and casing/diffuser.

$$\Delta P_{loc} = k_6 \rho\omega^2 {D_2}^2 \left(\frac{Q}{\omega {D_2}^3}\right)^2 \tag{17}$$

The localized head losses are written according to the dimensional analysis as follows:

$$C_{H_{loc}} = k_6 {C_Q}^2 \tag{18}$$

The distortion losses represent the sum of the shock losses, losses due to secondary flow and recirculation losses.

$$\Delta P_{distortion} = \Delta P_{shock} + \Delta P_{sec.flow} + \Delta P_{recirculation} \tag{19}$$

The shock losses are inertial losses, caused by the misalignment of the fluid velocity with the vane surface, when the pump operates out of best efficiency point (BEP) of the maximum diameter.

$$\Delta P_{shock} = \rho\omega^2 {D_2}^2 k_4 \left[1 - k_5^* \frac{Q}{\omega {D_2}^3}\right]^2 \tag{20}$$

Where k_5^* is based on pump geometry and is defined as:

$$k_5^* = \frac{{D_2}^3 \cot \beta_1}{2\pi B_1 r_1^2} \tag{21}$$

The losses due to secondary flow are relevant at partload and, as a consequence, also depend on the flow rate. Secondary flow can be caused by difference between the pressure at the suction side and the pressure at the pressure side of the impeller vanes, especially in the region closer to the outlet of the vane.

The recirculation losses occur inside the impellers and depend on the flow rate. These losses are maximized on zero flow (Gülich, 2007) and are expected to be zero at the best efficiency point.

For a one-dimensional simple model, the effects of the recirculation losses, losses due to secondary flow and shock losses overlap themselves, which makes it difficult to separate them. In this model they are represented as just one type of loss, called distortion loss. Its equation is similar to the shock losses one, but without an explicit definition for k_5^*. Therefore, the distortion losses are defined as follows:

$$\Delta P_{distortion} = \rho\omega^2 {D_2}^2 k_4 \left[1 - k_5 \frac{Q}{\omega {D_2}^3}\right]^2 \tag{22}$$

And the distortion head losses are written as follows:

$$C_{H_{distortion}} = k_4 \left[1 - k_5 C_Q\right]^2 \tag{23}$$

Therefore, the differential pressure curve can be understood as follows:

$$\Delta P = \frac{1}{4}{D_2}^2\rho\omega^2 - \frac{Q\rho\omega k_1}{D_2} - \frac{Q^2\rho\left[\frac{D_2\mu k_2}{Q\rho} + \left(\frac{D_2\mu}{Q\rho}\right)^n k_3\right]}{{D_2}^4} - {D_2}^2\rho\omega^2 k_4\left(1 - \frac{Qk_5}{{D_2}^3\omega}\right)^2 - \frac{Q^2\rho k_6}{{D_2}^4} \tag{24}$$

According to the dimensional analysis, it follows that:

$$C_H = \frac{1}{4} - k_4 + (-k_1 - Xk_2 + 2k_4k_5)C_Q + \left[-\left(\frac{X}{C_Q}\right)^n k_3 - k_4k_5^2 - k_6\right]C_Q^2 \tag{25}$$

The interpretation of the head versus flow rate curve of a centrifugal pump requires the explicit definition of the coefficient k_1 from the pump geometry and the simultaneous adjustment of the other six non-dimensional parameters $(k_2, k_3, k_4, k_5, k_6, n)$.

When the fluid viscosity is low, the parameter X is very small and its influence on the head curve can be neglected. In this case, the pump performance depends only on the flow rate and pump geometry.

$$C_H = F\left(C_Q, \frac{L_i}{D_2}\right) \tag{26}$$

Therefore, the head curve is represented by the following:

$$C_H = \frac{1}{4} - k_4 + (-k_1 + 2k_4k_5)C_Q + [-k_4k_5^2 - k_6]C_Q^2 \tag{27}$$

4. TESTS AND PUMP TYPES

The tests used in this work were provided by a global pump manufacturer. In order to ensure the correspondence among the hydraulic dimensions of the pump, only performance tests at the impeller maximum diameter have been considered, so that the impeller outlet angle β_2 and the impeller outlet width B_2, for instance, would in fact correspond to the impeller outlet diameter, D_2. Furthermore, it was possible to select tests of several types of pumps. These pump types are presented here according to the classification presented by the international standard API 610 11th edition, "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries".

The OH2 pumps are horizontal, centerline-mounted, single-stage overhung pumps, with single suction. They are widely used in off-shore and on-shore applications and also in the general industry. Twenty-one tests of OH2 pumps were analyzed in this work, with a range of dimensionless specific speeds from 0.11 to 0.83.

The BB1 pumps are horizontal, between bearings, axial split, volute pumps, with double suction impeller. According to the API 610 11th edition, this type of pump has one or two stages, but in this work just the pumps with one stage are considered in the category. These pumps are used especially in pipeline applications, refineries and in the water and wastewater markets. Seventeen tests of BB1 pumps were considered in this work, with a range of dimensionless specific speeds from 0.31 to 1.35. These pumps have a larger impeller outlet width B_2, that corresponds to the two sides of the impeller, which were used in the calculations.

The BB2 pumps are horizontal, between bearings, radial split volute pumps with a double suction impeller. According to the API 610 11th edition, this type of pump has one or two stages, but in this work just the pumps with one stage are considered in the category. This pump type is used in off-shore applications, such as SRU pumps, and in applications with high temperature. In this work, two tests of BB2 pumps were considered, with dimensionless specific speeds of 0.32 and 0.47. Like the BB1 pumps, the B_2 used in the calculations corresponds to both sides of the impeller.

Similar to the BB1 pumps, the BB3 are horizontal, between bearings, axial split volute pumps, but with multiple stages. In this work the pumps with two stages or more are considered in this category. These pumps are also used in pipeline applications and in the water and wastewater markets, but in applications with a higher flow rate and pressure than the ones in which the BB1 pumps are used. In this work, nine tests of BB3 pumps were analyzed, with a range of dimensionless specific speeds from 0.49 to 0.92. Like the BB1 and BB2 pumps, the B_2 used in the calculations corresponds to both sides of the impeller. Furthermore, the hydraulics were analyzed considering just one impeller. The pressure provided by the pump was divided by the number of stages of the pump.

The BB4 and BB5 pumps are horizontal, between bearings, radial split pumps, with diffuser and multiple stages. BB4 pumps have a single-casing, while the BB5 ones have a double casing (barrel). BB4 pumps are mainly used as boiler feed water pumps in the power market. BB5 pumps support the highest pressures and therefore are used in high reliability applications, such as injection pumps on platforms. Since the hydraulics of BB4 and BB5 pumps are usually the same, they were considered as a single group in this work. Twenty four tests were considered in this category. Like the BB3 pumps, the hydraulics were analyzed considering just one impeller. The pressure provided by the pump was divided by the number of stages of the pump. In this work, twenty four tests of BB4-BB5 pumps were analyzed, with a range of dimensionless specific speeds from 0.24 to 0.50.

Wet pit, vertically suspended, single-casing volute pumps with discharge through the column are designated as VS2 pumps. These pumps are widely used in off-shore application, such as fire-fighting system and as seawater lift pumps. They are also used in the water and wastewater market. Seven tests of VS2 pumps were analyzed in this work, with a range of dimensionless specific speeds from 0.89 to 1.49. It is important to note that, for vertical pumps, at the differential pressure calculation, the level difference between the bowl and the discharge nozzle has to be taken into account, as well as the pressure losses at the column pipe.

In total, eighty tests were analyzed in this work, with a range of dimensionless specific speeds from 0.11 to 1.49. Six different types of pump were considered, including volute and diffuser pumps, single and double suction, horizontal and vertical, single or multistage. This diversity on the tests is essential for the validation of the proposed model.

5. RESULTS AND DISCUSSION

Initially, the tests used in this work are presented. They are compared to the reference curves provided by the pump manufacturer, which are henceforth referred to as theoretical curves. This comparison is not shown for all the eighty tests considered in the current work, just for two tests of each pump type.

Then, the tested curves are compared to the ones adjusted by the model presented by Biazussi (2014). Again, this result is shown just for some tests.

After that, the model equation coefficients are plotted against the pump specific speed in order to analyze the correlation among them. From the analysis of these correlations, equations are proposed, so that the coefficients can be defined from basic characteristics of the pumps, such as specific speed and main dimensions. As a result, the head curve against flow can be defined, only based on these characteristics.

Finally, the correlation based curves are compared to the tested curves and the difference between them is discussed.

5.1 Tested versus theoretical curves

Figure 1 presents a selection of tested flow versus head curves over the theoretical ones provided by the pump manufacturer.

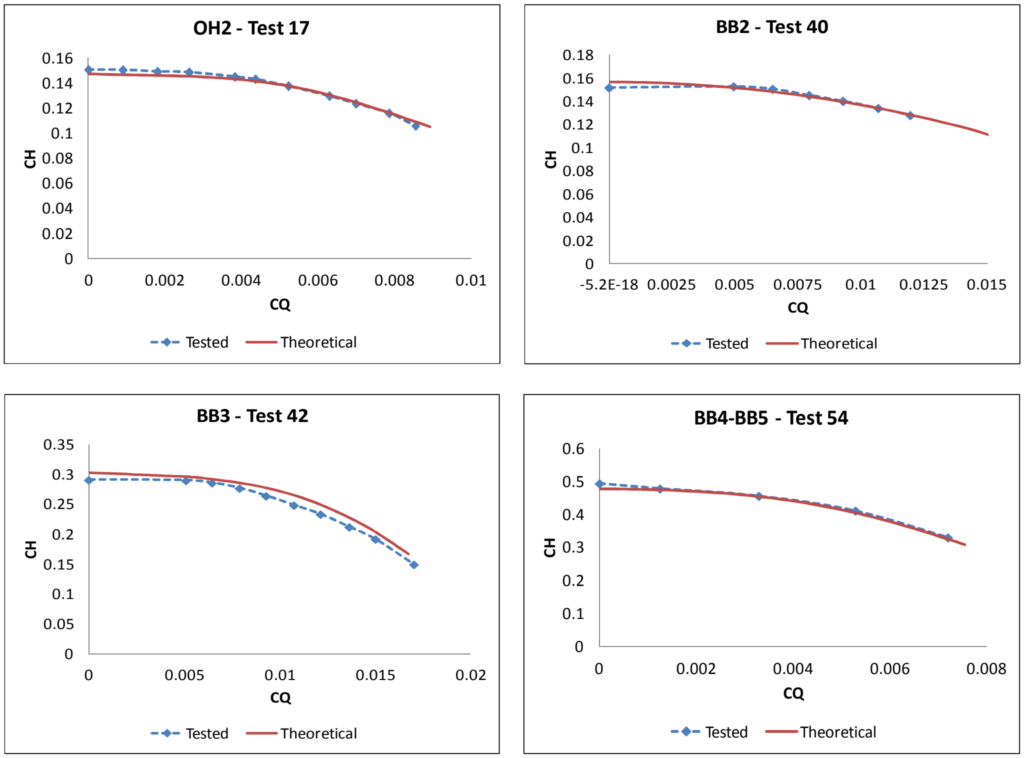

Figure 1: tested versus theoretical curves

Besides the experimental data error, the most relevant factors that could result in the differences observed among the curves are variations both in the casting process and during the test procedure. The test standards already allow for certain tolerances in the test results. So there is a margin to accommodate these variations.

The curves considered as reference for the model adjustment and the analysis of the results are the tested ones, since they represent the real performance of the pumps considered in this work.

5.2 Model adjustment for the head curve

The model adjustment proposed in Eq. (27) was done by the software Wolfram Mathematica ™ version 9.0.1. The "FindFit" function was used to find the numerical values of the equation coefficients that best fit the input data. The adjustment is based on the minimization of the norm, calculated as follows:

$$Norm = \sqrt{Abs(x)^2 + Abs(y)^2 + Abs(z)^2 + \cdots + Abs(n)^2} \tag{28}$$

Where n is the quantity of input data and the variables x, y and z are the deviation between the tested and the calculated values. The coefficient k_1 is defined based on the pump geometry, according to Eq. (11). The coefficients k_4, k_5 and k_6 are adjusted based on the tested data. In this work, calculated curves are the curves adjusted by the model. Figure 2 presents the calculated curves over a selection of tested curves.

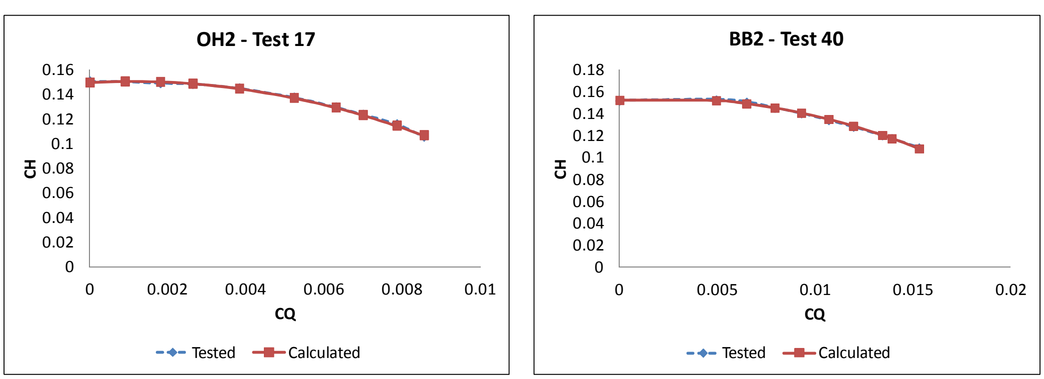

Figure 2: tested versus calculated curves

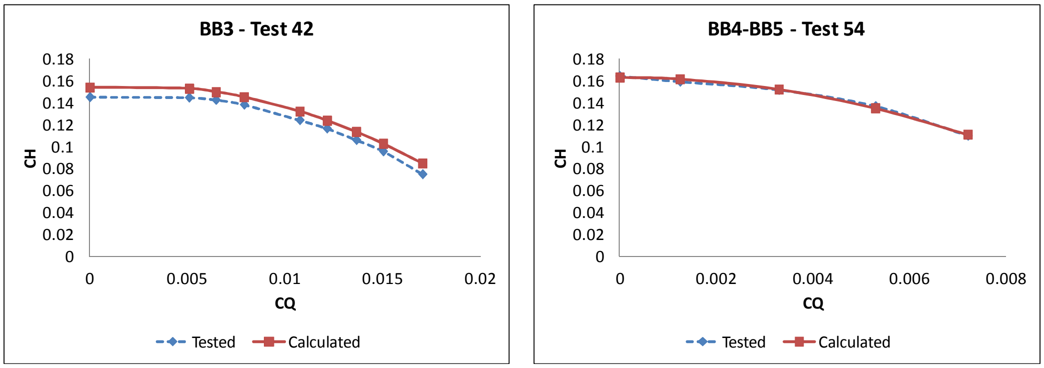

Figure 2: tested versus calculated curves (continued)

5.3 Model equation coefficients versus pump geometry

The coefficients k_4, k_5 and k_6 are also dependent on the pump geometry and can be represented as a function of the dimensionless specific speed. This analysis was done by Biazussi (2014), but only with a few test results, and the correlation found among the coefficients and the dimensionless specific speed was a straight line. Biazussi (2014) has even suggested that more tested data should be used in order to better understand these correlations. And this is the biggest contribution of the current work. With the data from eighty tests, it was possible to evaluate these correlations and, in fact, they proved not to follow straight lines.

In Fig. 3, Fig. 4 and Fig. 5, the equation coefficients k_4, k_5 and k_6 adjusted by the model for all the tests are plotted as functions of the dimensionless specific speed. The points are grouped according to the pump type, but it is possible to notice tendencies when analyzing the results of all the pump types together.

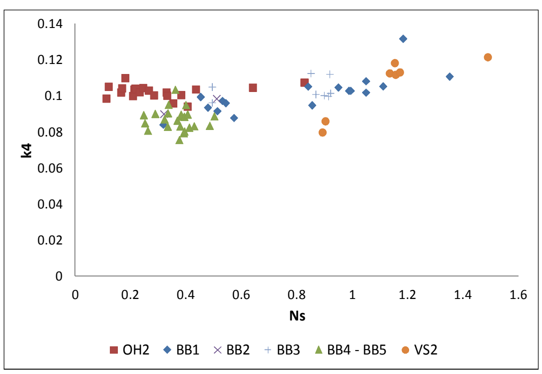

Figure 3: coefficient *k4* versus *Ns*

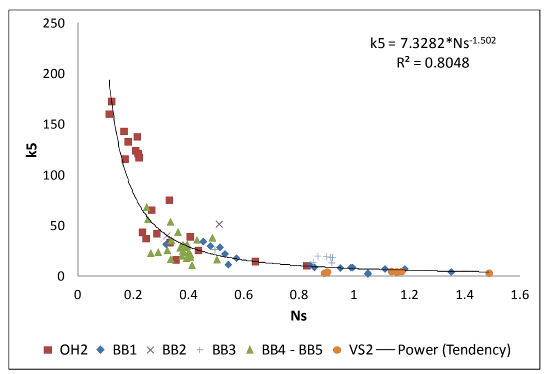

Figure 4: coefficient *k5* versus *Ns*

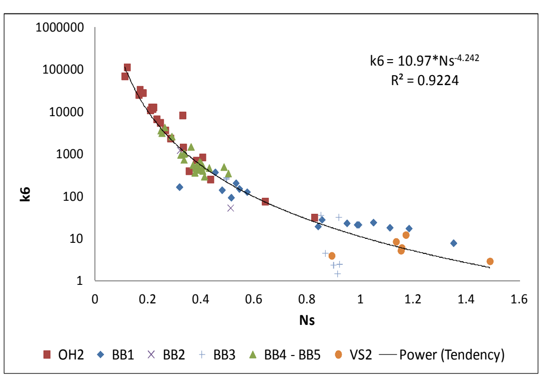

Figure 5: coefficient *k6* versus *Ns*

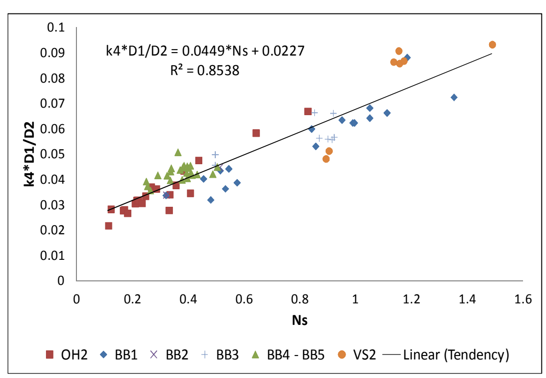

Figure 6: *k4*D1/D2* versus *Ns*

The coefficients k_5 and k_6 could be represented as functions of the dimensionless specific speed directly. The value of the coefficient k_4 is around 0.1 for most tests. However, in order to define a refined correlation for this coefficient, the ratio between the impeller inlet and outlet diameters, D_1 and D_2, has to be also taken into account. A reasonable correlation has been found between $k_4 \frac{D_1}{D_2}$ and the specific speed. Figure 6 shows this correlation.

This ratio $\frac{D_1}{D_2}$ is purely geometric and represents the effect of the impeller vane length on the differential pressure. If the impeller inlet and outlet diameter were the same, no differential pressure would be observed.

The correlations between the coefficients and the dimensionless specific speed can also be represented as equations that are the best fit for each group of points. They are shown as the tendency lines in Fig. 3, Fig. 4, Fig. 5 and Fig. 6.

With these correlations and Eq. 11 and Eq. 27, the head curve against flow can be defined only based on basic characteristics of the pump, namely D_2, β_2, B_2, D_1 and N_s, as shown in Table 1:

Table 1: correlations among the equation coefficients and basic information of the pump

$$C_H = \frac{1}{4} - k_4 + (-k_1 + 2k_4k_5)C_Q + [-k_4k_5^2 - k_6]C_Q^2 \quad (27)$$

Where:

$$k_1 = \frac{D_2 \cot \beta_2}{2\pi B_2} \quad (11)$$

$$k_4 = (0.0449N_s + 0.0227)\frac{D_2}{D_1} \quad (29)$$

$$k_5 = 7.3282{N_s}^{-1,502} \quad (30)$$

$$k_6 = 10.97N_s^{-4,242} \quad (31)$$

In this work, correlation based curves are the head versus flow curves based on equations presented in Table 1. Figure 7 compares the correlation based curves (Table 1) to the testes ones.

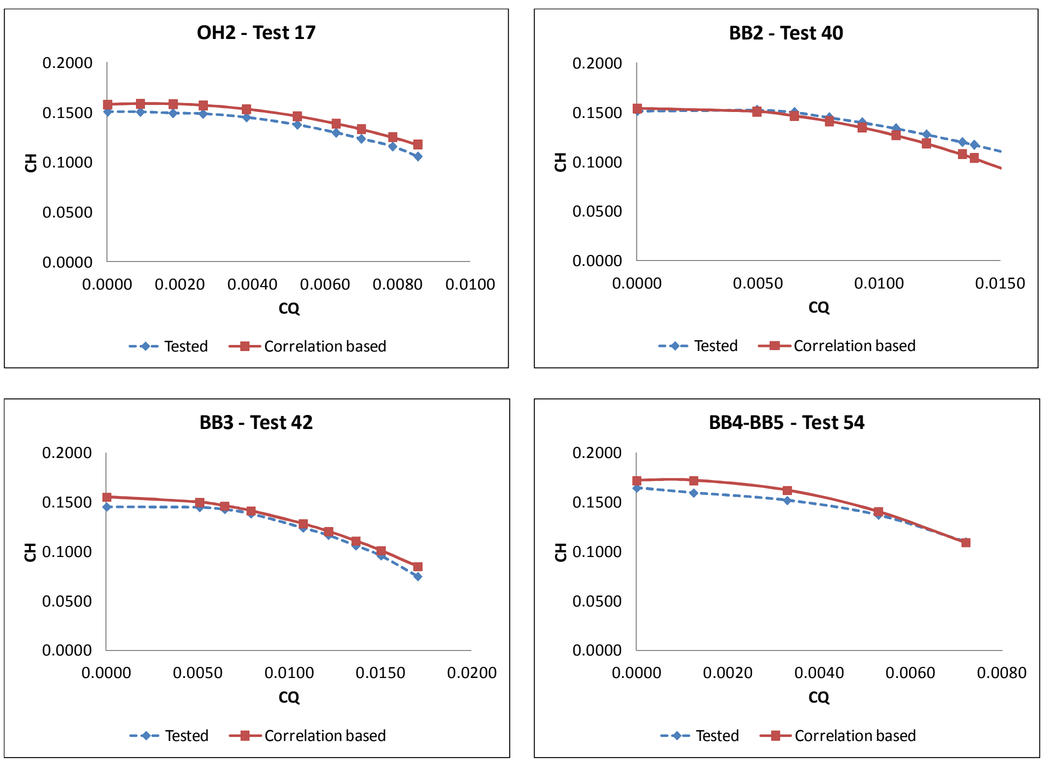

Figure 7: tested versus correlation based curves

In order to compare the correlation based curves to the tested ones, the standard deviation was calculated. By analyzing all the comparisons, three different behaviors are observed.

In most cases (64 tests), the correlations (Table 1) were able to predict the tested curve with good accuracy. The standard deviation limit set to define good accuracy is 0.02. This behavior was observed in all the pump types.

In some cases (10 tests), the correlations (Table 1) were able to predict the shape of the tested curve, but an offset was observed due an error on the shut-off head prediction. This behavior was observed in few cases of all pump types.

In few cases (6 tests), the shape of the curve was not well predicted. This behavior was observed in 2 tests of BB1 pumps and 4 tests of BB3 pumps. It happened more critically with the tests of BB3 pumps, but it is important to note that the pump hydraulics was the same in these 4 tests.

6. CONCLUSION

In literature, there are several models that attempt to predict the hydraulic performance of centrifugal pumps. Some of them are based on fluid dynamic principles, but others are based on empirical data and there are also the modern ones based on numerical simulation. In order to decide the best one to be used, the available information about the pump has to be taken into account.

The proposed correlations represent the mean tendency among a huge amount of experimental data from several types of pumps. Therefore the provided values of the coefficients may not be so accurate in some cases.

Given all the assumptions and simplifications, the objective of this work is to present correlations applicable to several pump types that easily provide a prediction of the head curve with reasonable error. This error is natural and expected. Since pumps have numerous configurations and their hydraulics are very complex, it is not expected that their performance be accurately predicted with only a handful of parameters.

Therefore, considering the goal of this work and all the tested data used to validate it, it is possible to affirm that the proposed correlations (Table 1) are able to easily predict the head curve of pumps in several different configurations with reasonable error.

7. REFERENCES

API 610, 11th edition, 2009. Centrifugal pumps for petroleum, petrochemical and natural gas industries. Winterthur: ISO copyright office.

Asuaje, M., Bakir, F., Kouidri, S., Kenyery, F., Rey, R., 2005. "Numerical modelization of the flow in centrifugal pump: volute influence in velocity and pressure fields." International Journal of Rotating Machinery.

Biazussi, J. L., 2014. “Modelo de deslizamento para escoamento gás-líquido em bomba centrífuga submersa operando com líquido de baixa viscosidade.” Ph.D. Thesis. Curso de Ciências e Engenharia de Petróleo, Departamento de Engenharia do Petróleo, Universidade Estadual de Campinas. Campinas.

Biazussi, J. L., Verde, W. M., Varón, M. P., Bannwart, A. C., 2013. "Comparison of experimental curves of ESP with a simple single-phase approach." 22nd International Congress of Mechanical Engineering (COBEM 2013). Ribeirão Preto.

Cheah, K.W., Lee, T. S., Winoto, S. H., Zhao, Z. M., 2007. "Numerical flow simulation in a centrifugal pump at design and off-design conditions." International Journal of Rotating Machinery.

Das, L. G., Sen, P. K., Saha, T. K., Chanda, A., 2010. "Performance prediction of vertical submersible centrifugal slurry pumps." International Journal of Fluid Machinery and Systems.

El-Naggar, M. A., 2013. "A one-dimensional flow analysis for the prediction of centrifugal pump performance characteristics." International Journal of Rotating Machinery.

Gülich, J. F., 2007. Centrifugal pumps, 2nd edition. Berlin: Springer.

Jafarzadeh, B., Hajari, A., Alishahi, M. M., Akbari, M. H., 2011. "The flow simulation of a low-specific-speed high-speed centrifugal pump." Elsevier.

Li, W. G., Su, F. Z., Xue, J. X., Xiao, C., 2002. "Experimental investigations into the performance of a commercial centrifugal oil pump." World Pumps.

Patel, D. P., Srivastava, R. K. and Shah, C. S., 1981. "Performance prediction in complete range of centrifugal pumps." 17th Technical Conference of British Pump Manufacturers' Association.

Paternost, G. M., 2013. "Estudo experimental sobre bomba centrífuga operando com fluido viscoso e escoamento bifásico gás-líquido." Dissertação de Mestrado. Curso de Ciências e Engenharia de Petróleo, Departamento de Engenharia do Petróleo, Universidade Estadual de Campinas. Campinas.

Shah, S. R., Jain, S. V., Patel, R. N., Lakhera, V. J., 2013. "CFD for centrifugal pumps: a review of the state-of-the-art." Elsevier.

Stepanoff, A. J., 1957. Centrifugal and axial flow pumps. New York: John Wiley&Sons.

Sun, D., Prado, M., 2003. "Single-phase model for ESP's head performance." SPE Production and Operations Symposium.

Sun, J., Tsukamoto, H., 2001. "Off-design performance prediction for diffuser pumps.".

Thin, K. C., Khaing, M. M., Aye, K. M., 2008. "Design and performance analysis of centrifugal pumps." World Academy of Science, Engineering and Technology.

8. RESPONSIBILITY NOTICE