Understanding Uncertainties in Viscous Performance Predictions for Centrifugal Pumps

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ABSTRACT

Accurate predictions of centrifugal pump performance are critical in every fluid industry. Many industries encounter situations where a fluid more viscous than water must be conveyed with a centrifugal pump. As liquid viscosity increases, losses in the pump increase and degrade its performance. Pumps are nearly always tested only with pure water, meaning some estimation of this performance degradation must be made for pump and motor selection. There are several ways to make this estimation, all of which carry uncertainty. A widely adopted method is described in *ANSI/HI 9.6.7-2015 – Rotodynamic Pumps – Guideline for Effects of Liquid Viscosity on Performance*. Proper application and understanding of the uncertainties in HI 9.6.7-2015 are described here. The analysis herein shows that recent test data falls within reasonable uncertainty levels, and recommendations for pump and motor sizing accounting for these uncertainties are

given. Finally, the HI 9.6.7-2015 method correlations are based on trustworthy but limited data – a plea to the industry is made to provide the Hydraulic Institute with additional viscous pump measurements for the purposes of continually improving these correlations.

INTRODUCTION

CFD analysis or other detailed breakdowns of the viscous losses in centrifugal pumps are possible but often not feasible due to cost and complexity. Instead, relatively simplified predictions can be made based on quantities likely at hand; water performance curves, impeller rotational speed, kinematic viscosity and specific gravity. The HI 9.6.7-2015 guideline is one such method, but it is an empirical method and therefore carries with it some uncertainty.

Recent publications showing tested viscous performance differing from the HI 9.6.7-2015 predictions may cause concern over the accuracy of the prediction. Several studies were examined in detail – two of which explicitly compared measured viscous performance to the HI 9.6.7-2015 predictions.

- A laboratory test of a two-stage vertically suspended pump, by Le Fur, et al. (2015) at CETIM, published at the 44th Turbomachinery & 31st Pump Symposia, which explicitly compares measured viscous performance to HI predictions.
- A laboratory test of a single-stage double suction between bearings pipe line style pump, by Robinett (2017) on the behalf of the Pipeline Research Council International (PRCI), which was focused on examining the effect of fluid viscosity on pump performance, especially as it relates to suction piping configuration.
- Field tests of single-stage between bearing pumps in series by Robinett and Fulghum (2019) on behalf of PRCI, which compared the results to HI 9.6.7-2015.
- Finally, an older study of two smaller pumps under viscous flow, by Ippen (1945) and published by ASME, is included to contrast against the recently collected data.

This paper seeks to clarify the uncertainties present in viscous predictions in general and in the HI 9.6.7-2015 guideline in particular. These uncertainties fall broadly into three categories: 1) the use of a single dimensionless number to characterize a complex phenomenon, 2) the limited data set used to construct the empirical model, and 3) uncertainties in field measurements or device characteristics.

These uncertainties cannot be eliminated without moving away from a simple empirical model. It is reasonable to make the assertion that different centrifugal pumps at different operating conditions may give the same base dimensionless parameter. However, it would be expected that two different pumps will have somewhat different performance. Therefore, even with perfect field measurements from a large quantity of centrifugal pumps, any simple empirical model is expected to show potentially significant uncertainty.

Quantifying this uncertainty is then of critical importance for such an empirical model to be useful. The intent herein is not to suggest an improvement to HI 9.6.7-2015, but instead only to provide users of the guideline an estimation of its precision. The results from the above studies are compared to the existing data set, and as expected the HI 9.6.7-2015 method does not exactly predict the performance, though it is clearly shown that the predictions are within what can be reasonably expected from the current model.

With a thorough understanding of uncertainty, centrifugal pumps and motors for viscous application can be sized with more confidence. As performance degrades, the uncertainty on predictions grows – for extreme or critical applications, it is prudent to use a conservative estimate of performance degradation based on the uncertainty in the predictions.

CHARACTERIZING A REDUCED CENTRIFUGAL PUMP PERFORMANCE CURVE

Reduced performance is typically characterized with correction factors on flow, head, and efficiency. This approach is also used to quantify reductions in performance in situations other than viscous pumping, such as in slurry pumping (ANSI/HI 12.1-12.6-2016).

The correction factors are simply direct multiplicative factors on an operating point described by the water performance curves, shown in Equations (1-3) and demonstrated for head in Figure 1. Efficiency corrections follow a very similar process.

$$C_Q = \frac{Q_{Viscous}}{Q_{Water}} \tag{1}$$

$$C_H = \frac{H_{Viscous}}{H_{Water}} \tag{2}$$

$$C_\eta = \frac{\eta_{Viscous}}{\eta_{Water}} \tag{3}$$

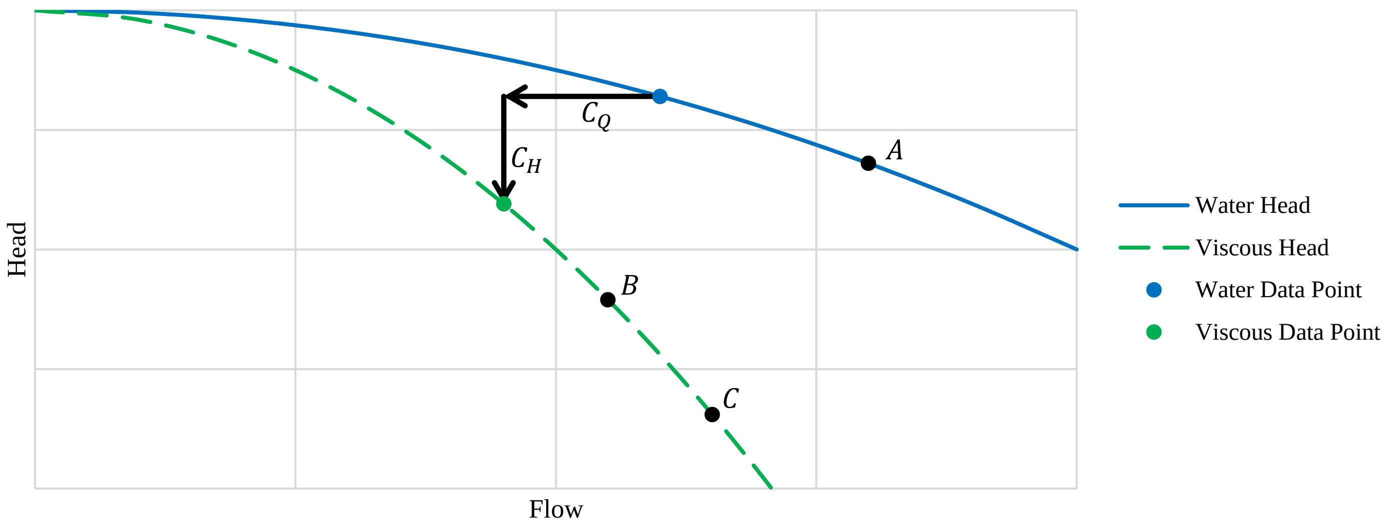

Figure 1. Simple example of correction factors.

It is important to recognize that these correction factors relate two operating points, not two curves. Without specific information about the test points making up two given curves, care must be taken in approximating the correction factors. For example, given some point A on the water curve in Figure 1, the corresponding viscous operating point might be B or C. Both options are possible, only requiring a different set of correction factors.

To generate a unique solution to this problem, additional constraints must be defined. This can be accomplished in part by holding the flow correction factor constant for all flows. This simplifies the problem greatly and forces the variability to be accounted for with the other correction factors.

Second, a relation between the flow and head corrections must be defined. Without this step, a flow correction factor can be chosen arbitrarily. Several options exist, and some possibilities have been presented here – none are necessarily more correct than any other.

- Option 1 – Define the flow correction factor as the ratio of viscous BEP flow to water BEP flow. This has an advantage of imparting physical meaning to the flow correction factor.
- Option 2 – Use the volute or diffuser characteristic – a straight line that passes through the origin of the head-flow plane and the water BEP. The flow correction factor would be a ratio between the viscous operating flow on the volute characteristic and the water BEP flow.
- Option 3 – Use a different function to determine the flow correction factor, based off the pump characteristics. This is, in effect, a predictive correction factor – one such function is the empirical HI 9.6.7-2015 prediction.

Herein, Option 3 has been selected. At first glance, it may seem odd to choose what could be described as a predictive correction factor in describing measured behavior. However, it is important to note that choosing this option does not equate to the imposition of physical restrictions on the model – as long as the head and efficiency correction factors are allowed to vary, any measured curve can still be reproduced exactly.

To effectively compare test correction factors to predicted correction factors, the calculation of both must follow the same form. Option 3 allows this direct comparison even though it does not have physical meaning like a ratio of BEPs. Because the flow correction factor will be considered a function of pump characteristics only, it will be identical in all cases for test and predicted coefficients. This simplifies greatly the prospect of determining the accuracy of the remaining coefficients.

VISCOUS PERFORMANCE DEGRADATION

It is helpful to briefly discuss why performance is degraded when pumping a more viscous fluid, as shown in Figure 2. While it is intuitive that a "thicker" fluid requires more energy to move, the details of how energy is lost (or in some ways, gained), are not necessarily immediately clear. Some of these effects are discussed here.

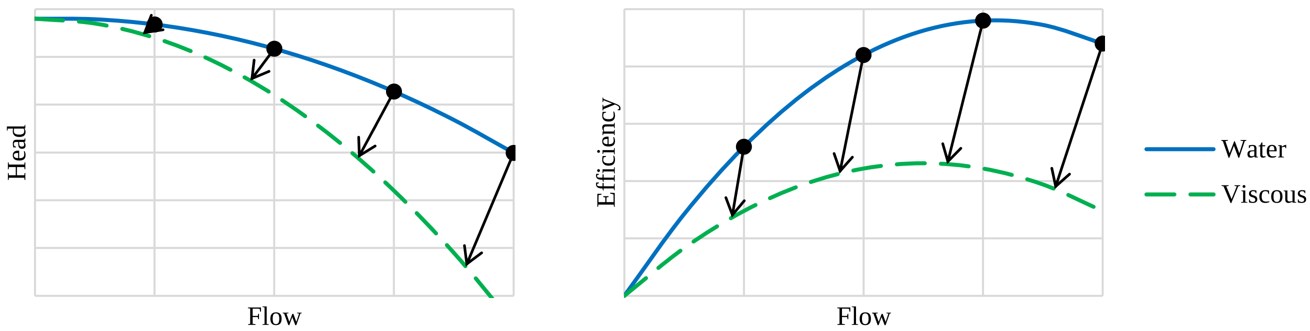

Figure 2. Impacts of viscosity on centrifugal pump performance.

Hydraulic Losses

Any flow of a real fluid past a fixed object will experience viscous losses. This is perhaps most intuitive for a simple pipe – for the same flow velocity, an increase in viscosity will decrease the Reynolds Number. This will in turn increase the friction factor and ultimately the head loss in the pipe.

Flow inside a pump is no different – an increase in viscosity increases friction and therefore head loss through the pump (Li 2013). This is quite difficult to accurately quantify, as geometry and fluid behavior changes dramatically through the pump. Fortunately, these hydraulic losses are strongly linked to a Reynolds Number – just as in a pipe. Like most models based on a single value, using a Reynolds Number imparts some estimation on the complex losses.

Disk Friction

Shear forces on the fluid between the rotating impeller and stationary pump casing generate a resistive torque, causing disk friction losses. The magnitude of this torque depends in part on the Reynolds Number – an increase in viscosity directly corresponds to an increase in power consumption. Disk friction is often a primary driver in pump efficiency and is therefore a major consideration in determining viscous performance. This effect is generally stronger for pumps of lower specific speed, and consequently these require more power in viscous operation (Daqiqshirazi et al. 2014). Pumps with specific speed of 500 US Units (10 metric) may lose 50% of their power to disk friction at low viscosities (1 cSt), whereas pumps with specific speed near 2300 US Units (45 metric) may lose 50% of their power to disk friction at high viscosities (3000 cSt) (ANSI/HI 9.6.7-2015).

The Reynolds Number and specific speed together can be used to make an estimation of the disk friction – these values carry with them an indication of the size and type of pump, speed of the impeller, and flows seen in the pump, all of which affect the disk friction. Again, it may not be reasonable to expect an extremely accurate estimation of this complex phenomenon from only two dimensionless parameters.

Volumetric Efficiency

Leakage losses under a viscous flow are likely to decrease due to the increased resistance to flow through small passages. This will in turn increase the efficiency of the pump somewhat, perhaps even enough to improve overall performance in moderate cases (Ippen 1945). Detailed analysis of this effect is particularly difficult due to the complex dynamics, geometry, temperature gradients, and fluid properties in these regions.

Thermal Effects

As the fluid experiences any type of friction, it will tend to heat up. This will change the fluid viscosity which impacts all the above losses. It can be seen from one of the studies analyzed here (Le Fur, et. al. 2015) that a viscous pumping system behaves very differently in a cold start versus steady operation. This clearly indicates a performance dependence on the fluid temperature in the pump.

METHODS FOR PREDICTING PERFORMANCE DEGRADATION

CFD Analysis

With the advancement of ever increasingly reliable numerical models for the description of complex flow phenomena in centrifugal pumps, the prediction of pump performance under higher viscosities becomes more and more feasible.

Such analysis is rather complex and requires properly resolved geometry, consideration of transient effects, and the capture of changing fluid viscosity and density due to thermal effects, which may be exacerbated by high viscosities. At very high viscosities, the flow inside the pump can be laminar, and specific models need to be used to describe the transition correctly. CFD analysis also allows for the analysis of local effects inside the pump, providing useful information for defining strategies to optimize the pump performance.

Though certainly more complex than a simple empirical model, much of the insight that can be gained from CFD analysis is very difficult to determine experimentally. In fact, as numerical codes and computers improve, the cost of such an analysis may be much lower than that of a specialized laboratory test. However, the barriers to CFD analysis are still high, and simpler, less accurate, methods are typically used to predict performance.

Detailed Loss Analysis

The individual losses explained in the section Viscous Performance Degradation can be used for a more detailed assessment of the influence of viscosity on pump performance. Gülich (1999) outlined a procedure for estimating the performance reduction using relatively simple correlations between head and disk friction losses and a limited number of operational and geometric parameters.

A more elaborate approach, requiring more in-depth information on the geometrical details of the analyzed pump usually only available to pump manufacturers, is to use empirically based calculations for all discussed individual losses (Gülich 2010). First, the internal pump geometry and rotational speed are used to estimate the theoretical (Euler) performance curve for water. The estimated viscous losses are then determined and applied to this theoretical curve, generating a predicted viscous curve.

While this approach is strictly only valid in an operating regime free of recirculation at the pump inlet and outlet, preliminary , ongoing analyses of available tests for one particular geometry by some of the present authors are promising and indicate a possible improvement of the prediction accuracy.

HI Method Prior to 2004 (ANSI/HI 1.1-1.5-1994)

The Hydraulic Institute updated the guideline in 2004 with the release of HI 9.6.7-2004. Prior to this, the process relied on a nomogram. Properly reading a nomogram is difficult and obtaining precise and repeatable numbers from one is near impossible. This was a major reason for the revision of the method, but not the only one. The older method had no direct dependence on specific speed and was overconservative in many cases. This drove the Hydraulic Institute to develop the improved method discussed here.

Current HI Method (ANSI/HI 9.6.7-2015)

The current method presented in HI 9.6.7-2015 is similar in approach to the old method, in that a dimensionless parameter is determined and subsequently used to find correction factors. Equations given by Gülich (2010) were adjusted to fit the data available to the HI 9.6.7 Committee at the time. The method is driven by the calculation of a dimensionless parameter B, which incorporates the effects of Reynolds Number and specific speed, as seen in Equation (4). A derivation for this parameter can be found in Appendix B. It is calculated at the water performance BEP.

$$B \propto \frac{\nu^{1/2}\Delta H^{1/16}}{Q^{3/8}N^{1/4}} \propto \frac{1}{{Re_B}^{1/2}{Ns}^{1/4}} \tag{4}$$

The HI 9.6.7-2015 method calculates a correction factor for flow per Equation (5) that is constant for all flows. This is Option 3 described in the section Characterizing a Reduced Centrifugal Performance Curve.

$$C_Q = f(B) = constant \tag{5}$$

Additionally, due to the strong relation between efficiency and disk friction losses discussed in the section Viscous Performance Degradation, it may be considered reasonable that the efficiency correction factor is constant for all flows for a given B, as shown in Equation (6). If the efficiency correction is considered constant, the viscous BEP will always be predicted to lay on the volute characteristic.

$$C_\eta = f(B) = constant \tag{6}$$

The HI 9.6.7-2015 guideline assumes shutoff head is not dependent on fluid viscosity. To force the head correction factor to be equal to one at zero flow, its calculation follows the form in Equation (7). Additionally, the original HI 9.6.7-2015 test data showed that the volute constraint is valid for viscous fluids – therefore, the flow and head correction factors are also forced to be equivalent at the water performance BEP.

$$C_H = 1 - \left[(1 - C_Q) \left(\frac{Q_{Water}}{Q_{BEP-W}} \right)^{0.75} \right] \quad (7)$$

The method specifies limitations on applicability; namely that the pump is of a conventional, radial type (Ns <= 3000 US Units, Nq <= 58 metric), the B parameter is below 40, and the fluid is Newtonian with viscosity between 1 and 4000 cSt.

DETERMINING TEST AND PREDICTION CORRECTION FACTORS

Test Correction Factors

There are two sets of points available, one from the measured water curve, and one from the measured viscous curve. These points do not directly correspond to one another. Therefore, the location of points on the water curve that result in the measured viscous data points must be estimated. These points on the water curve are estimated in two ways.

First, their flow is determined by assuming the HI 9.6.7-2015 flow correction factor is accurate, as discussed in the section Characterizing a Reduced Curve. Second, the water head and efficiency must be determined, but it is unlikely measured data exists at the desired flow, meaning interpolation of some type is required. If the water curve is composed of a reasonable number of data points this error is expected to be small.

To demonstrate the calculation of test correction factors, the measurement series PRCI17-E was selected (see Table 1 in Appendix A for a list of all tests). First C_{Q-HI} is calculated per HI 9.6.7-2015. Second, a viscous data point $(Q_{Viscous}, H_{Viscous})$ is selected. Third, the effective water flow is determined by $Q_{Water} = Q_{Viscous}/C_{Q-HI}$. Fourth, H_{Water} is found via linear interpolation. Higher order curve fits may be warranted in some situations but were not applied here. This is demonstrated in Figure 3.

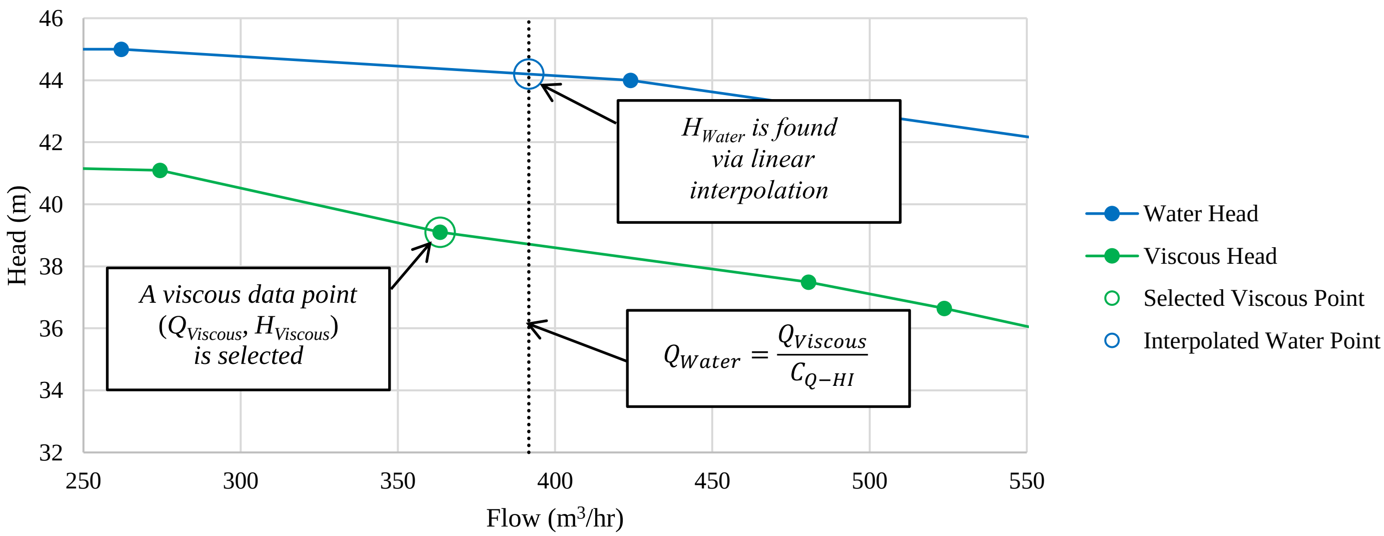

Figure 3. Determining an effective water head from viscous flow.

Prediction Correction Factors

Predicted correction factors that correspond to the test correction factors are determined from the interpolated water data points given the method outlined in HI 9.6.7-2015, as shown in Figure 4. Because C_{Q-HI} is used for both test and prediction, the points will only vary on the ordinate.

The test correction factors for head and efficiency will be simply the ratio of the tested viscous value to these interpolated values $(C_{H-Test} = H_{Viscous}/H_{Water})$.

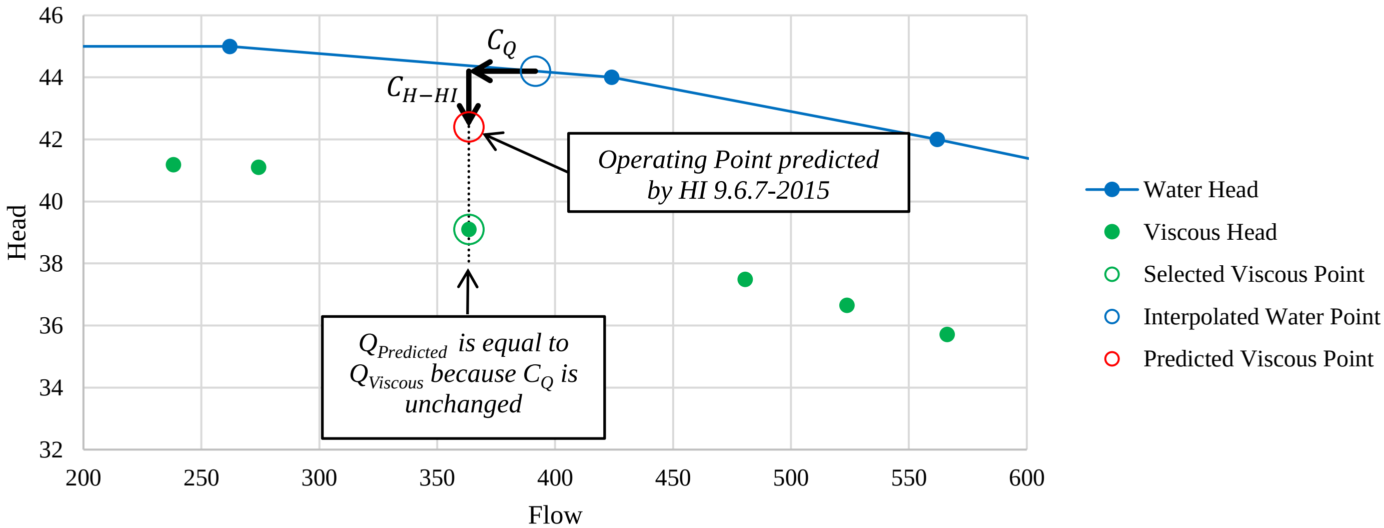

Figure 4. A predicted viscous data point using the same C_{Q-HI} varies only in head.

MEASURED VISCOUS PERFORMANCE

In total, seven pumps across the four studies were considered – this section contains brief discussions of those pumps and the trends of the results (see Table 1 in Appendix A for a list of all tests).

Vertical Pump Laboratory Study (CETIM15) (Le Fur, et. al. 2015)
A laboratory test of a large 2-stage vertical pump with a specific speed of 2611 US Units (51 metric). This study explicitly compared the test correction factors to those predicted by HI 9.6.7-2015 and HI 1.1-1.5-1994.

As shown in Figure 5, tested values for head agreed relatively closely with HI 9.6.7-2015 predictions, though the tested head values were higher than the prediction at lower flows – higher than the water curve in many cases. Tested values for power were substantially higher than the predictions, a trend that grew with increasing flow. Note that the series CETIM15-C shows an inconsistent trend on power. The measured water and viscous curves from Le Fur's study were used herein to determine the accuracy of the HI 9.6.7-2015 predictions, Le Fur's own calculations and conclusions on the predictions are very similar but were not incorporated directly.

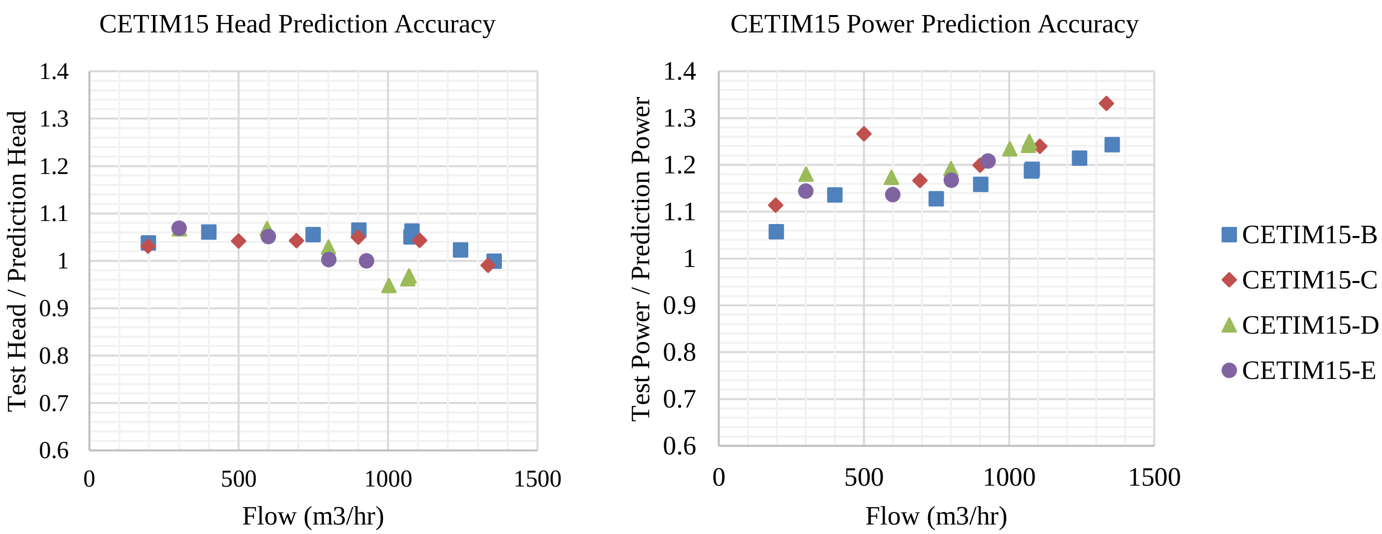

Figure 5. Head and Power prediction accuracies for the CETIM15 test series. Ns = 2611 US Units (51 metric)

Viscous Effects on Suction Piping Laboratory Study (PRCI17) (Robinett 2017)
A laboratory test of a moderately sized single stage double suction between bearings pump with a specific speed of 2435 US Units (47

metric). This study was designed to address the effects of suction piping layout and fluid viscosity on pump performance. One series of tests considers straight suction piping and this data is considered here.

Figure 6 shows that tested values for head were somewhat lower than HI 9.6.7-2015 predicted. For higher viscosities, tested power was lower than predicted, though the lowest viscosity test (PRCI17-B) showed the test power to be higher than predicted.

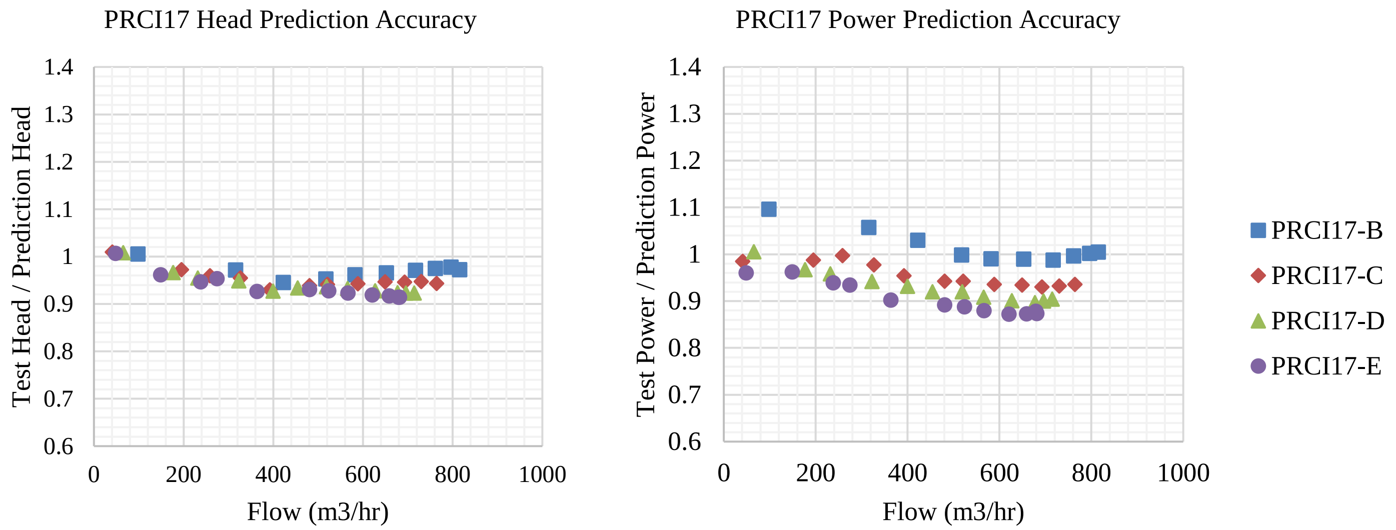

Figure 6. Head and Power prediction accuracies for the PRCI17 test series. Ns = 2435 US Units (47 metric)

Field Tests of Crude Oil Pipeline Pumps (PRCI19) (Robinett and Fulghum 2019)
This study considered three pumps of the same single stage double suction between bearings model, with a specific speed of 1671 US Units (32 metric). The first (PRCI19.1) and second (PRCI19.2) pumps are constant speed and operate with similar pumps configured in series. The third (PRCI19.3) is variable speed pump last in a similar series and was explicitly tested at different speeds. This study compared the test correction factors to those predicted by HI 9.6.7-2015.

These tests showed, in most cases, close agreement with the predictions, as indicated in Figure 7. Most tests showed slightly lower head than the prediction, and slightly higher power than the prediction.

One test series – PRCI19.3-B – showed substantially lower head than predicted. It should be noted that this test point was taken outside the pump normal operating range (beyond end of curve), meaning the prediction was based on an extrapolated water curve. This results in high uncertainty unrelated to the viscous prediction uncertainty, and the test point should therefore be disregarded.

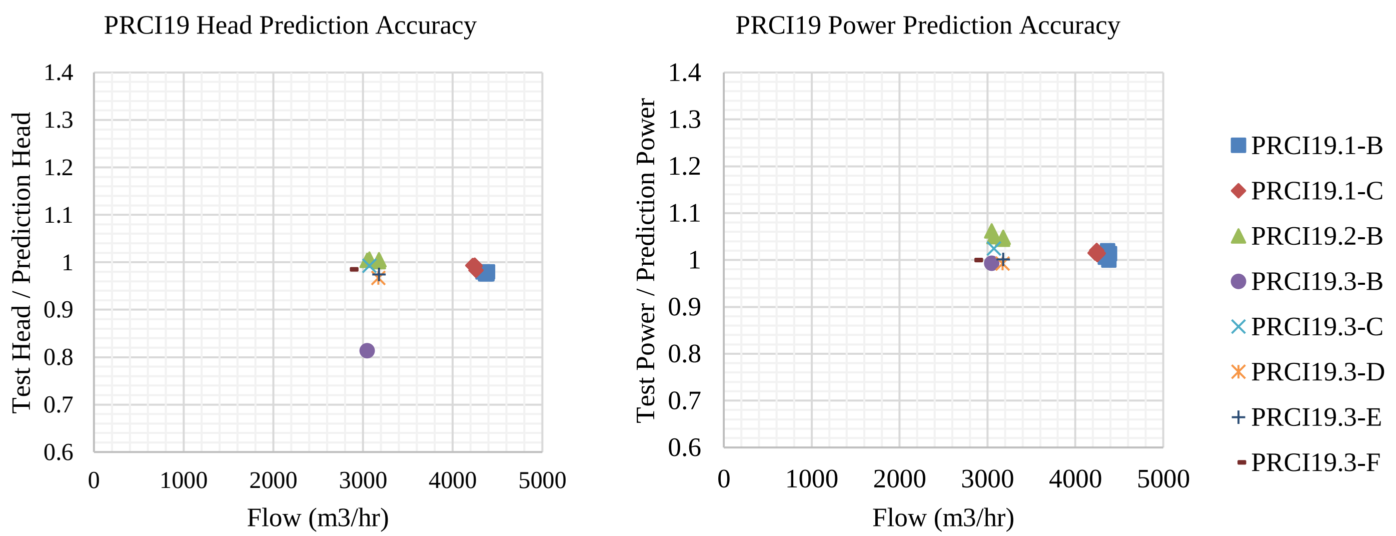

Figure 7. Head and Power prediction accuracies for all PRCI19 test series. Ns = 1671 US Units (32 metric)

Laboratory Tests of Smaller Pumps (IPPEN45) (Ippen 1945)
An older study referenced in HI 9.6.7-2015 – this study provided substantial data on the viscous performance of pumps smaller than the others considered here. The first (IPPEN45.1) was a moderately sized single stage double suction pump, with a specific speed of 2622 Us Units (51 metric). The second (IPPEN45.2) was a much smaller single stage single suction overhung pump with a specific speed of 1163 US Units (22.5 metric).

The first pump, seen in Figure 8, showed head results in line with the prediction, with the most viscous series (IPPEN45.1-G) showing higher head than predicted. Power was consistently tested higher than predicted, with the difference increasing toward lower flows.

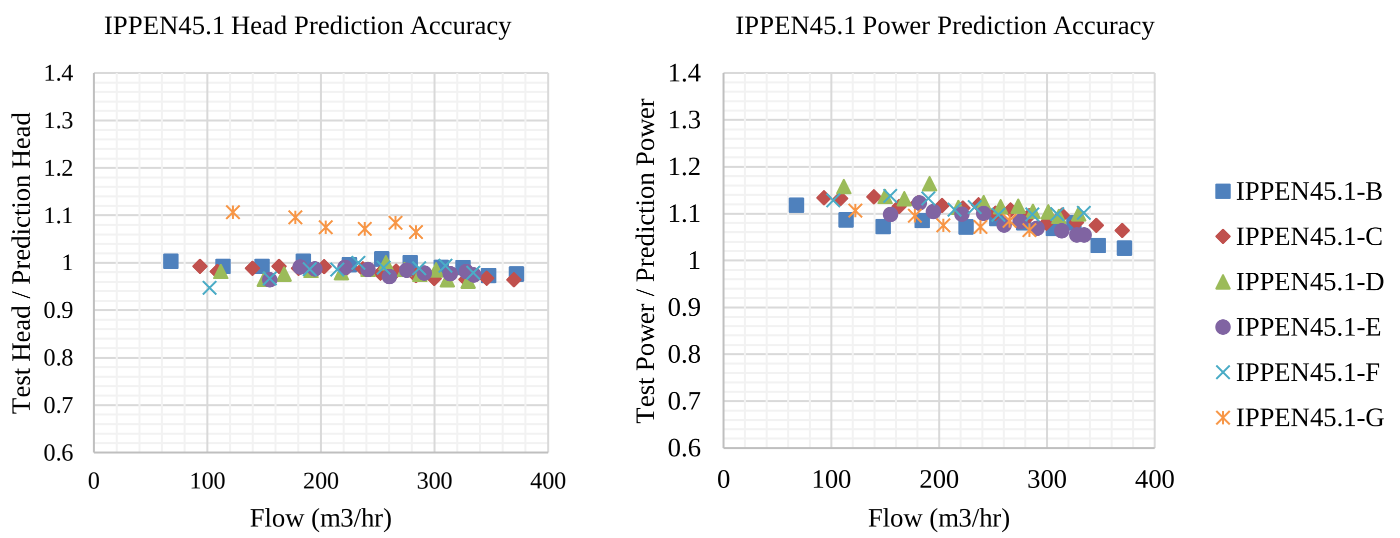

Figure 8. Head and Power prediction accuracies for all IPPEN45.1 test series. Ns = 2622 US Units (51 metric)

The second pump, seen in Figure 9, showed, by far, the most spread of any considered data set. The test results for head show, for lower viscosities, an increase in test head compared to prediction at increasing flow. However, the tested head is far lower than predicted for high viscosities, with a trend in the opposite direction. The series IPPEN45.2-H (855 cSt) has more than double the viscosity of IPPEN45.2-G (369 cSt). Power shows no obvious trends, other than the higher viscosities being, in general, less accurately predicted. Note that at these higher viscosities, the corrections are relatively extreme – for the IPPEN45.2-K series at 1877 cSt, the head correction factor drops below 0.5, and the efficiency correction factor is below 0.2. It is perhaps not unreasonable to expect greater error at these more extreme conditions.

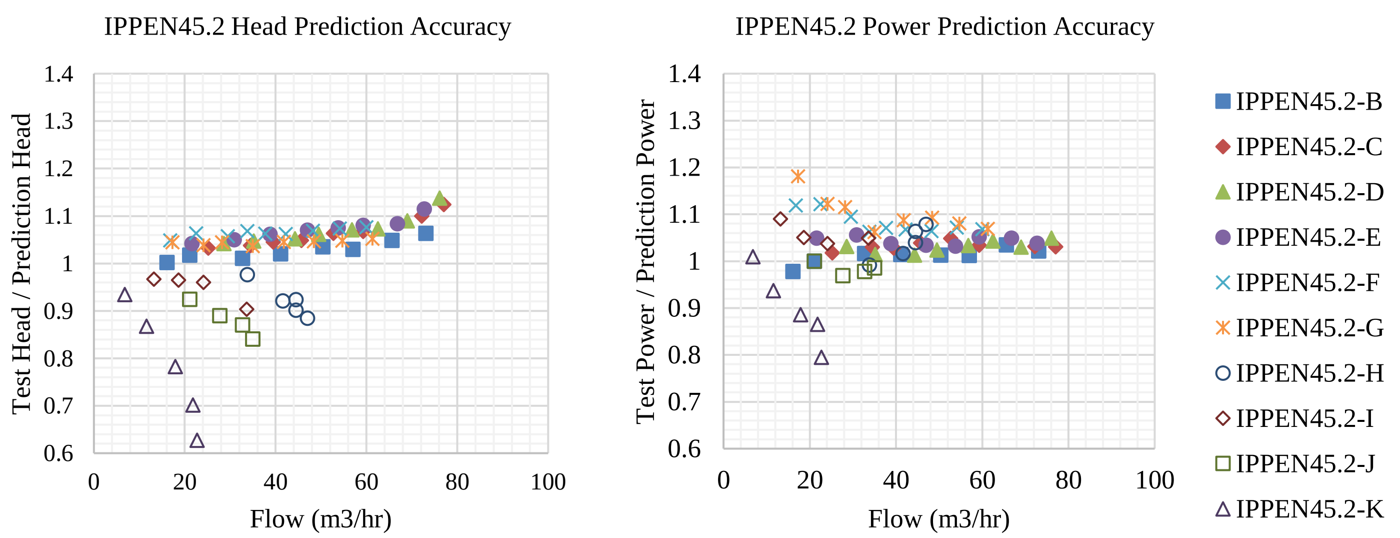

Figure 9. Head and Power prediction accuracies for all IPPEN45.2 test series. Ns = 1163 US Units (22.5 metric)

QUANTIFYING UNCERTAINTY

Recognizing Uncertainties in Measurement

No measurement is perfectly accurate and quantifying the uncertainty on a measurement is just as critical as the actual value. Uncertainty in measurement can come from many sources and can be difficult to quantify (Possolo 2015). Consider by example determining the efficiency of a pump with viscous fluids. Accurate fluid density, flowrate, developed head, and pump input power are all required.

- All instruments and instrument readings are subject to uncertainty.
- Fluid density within the pump casing might be impossible to read directly. The density might be estimated with fluid temperature and a property model.
- This temperature might be read some distance upstream or downstream of the pump, even though temperature can change substantially within the pump.
- Pump input power must be the actual power delivered to the pump. This could, for example, be mistaken as power required by the motor, or roughly estimated from motor power.
- All these uncertainties compound in ways that are difficult to quantify.

This paper is not intended to address these types of uncertainty, and all source data is presumed accurate. Regardless, it must be mentioned as the conclusions drawn by this type of analysis are impacted by these uncertainties. Even if the HI 9.6.7-2015 model was theoretically perfect, some deviation between test and prediction would be expected due to the above. For general discussion on accounting for uncertainties in test facilities, see ANSI/HI 14.6-2016 or ISO 9906-2012.

Quantifying Confidence in an Empirical Model

Ratios of Known to Estimated

There are many ways that the error between a known and estimated value might be represented. A simple method is to ratio the two values (Equation 8).

$$Error\ on\ Factor = \frac{X_{Known}}{X_{Estimated}} \tag{8}$$

This is the approach taken in the Measured Viscous Performance section – it is easy to calculate and intuitive to state "the test value is 10% higher than the prediction." Unfortunately, it has a significant drawback when quantifying the estimated uncertainty on a prediction.

Consider a predicted head correction factor of 0.90, on a water head of 100 meters. This would correspond to a predicted viscous head of 90 meters. The scatter on test data might imply that most data falls within 10% of the prediction. Using this as percentage error on an estimate, it could be claimed that the true viscous head is 90 meters $\pm$ 10%, or within the interval [81,99].

However, what if the fluid viscosity is increased – for the same pump – until the predicted head correction factor is 0.50? Applying the same logic, this would result in the statement that the true viscous head is 50 meters $\pm$ 10% or within the interval [45,55]. The range has decreased from 18 meters to 10 meters, implying higher accuracy even though the situation is more extreme.

Another approach might be to use a relative error (Equation 9), which will tend to show a larger error as the test correction factor goes to zero. This is desired behavior as a stronger correction is less certain, but it is still a ratio and presents the same problem as before.

$$Error\ on\ Factor = \frac{X_{Known} - X_{Estimated}}{X_{Estimated}} \tag{9}$$

Prediction Interval

The problem lies, in part, with attempting to apply a fixed interval to the problem as an uncertainty. It is desirable to state "the prediction is accurate to within $\pm$10%," but, as described above, this does not allow the uncertainty to vary with the value. To retain the effect of increasing uncertainty as the correction factor gets closer to zero, the interval must be allowed to change.

Additionally, statements such as "the prediction is accurate to within $\pm$10%" may imply an absolute certainty that the true value lies within the bounds. However, for an empirical model, there is *no* band that can be indicated to have, with absolute certainty, the true value. All values derived from empirical models are estimates. Therefore, the *probability* that the true value falls within the specified band must be found.

This is known as a *prediction interval* – a band around the empirical estimate that has a certain percentage chance of containing the actual value. It is not possible to bound the estimate with certainty, but it is possible to state that a true value has some probability of being within a given interval.

Applying Prediction Intervals to the HI 9.6.7-2015 Basis Data

The HI 9.6.7-2015 guideline is based on data gathered from a variety of pumps under various conditions (Gülich 2000). The test data considered:

- Single and multi-stage pumps
- Closed and open impellers
- Specific speeds from 310 to 2330 US Units (6 to 45 metric)
- Kinematic Viscosity from 1 to 3000 cSt
- Impeller diameters from 5.5 to 16 inches (140 to 406 mm)
- Water BEP flow from 32 to 1230 gpm (7.2 to 280 m^3/hr)
- Water BEP head from 30 to 427 feet (9 to 130 m)
- Water BEP efficiency from 28 to 86%

This data (points) is shown in Figure 10 with the HI 9.6.7-2015 model (solid line) overlaid. C_{H-Test} and $C_{\eta-Test}$ are measured correction factors from water performance per Equations (1-3). The B Parameter is determined per Equation (4).

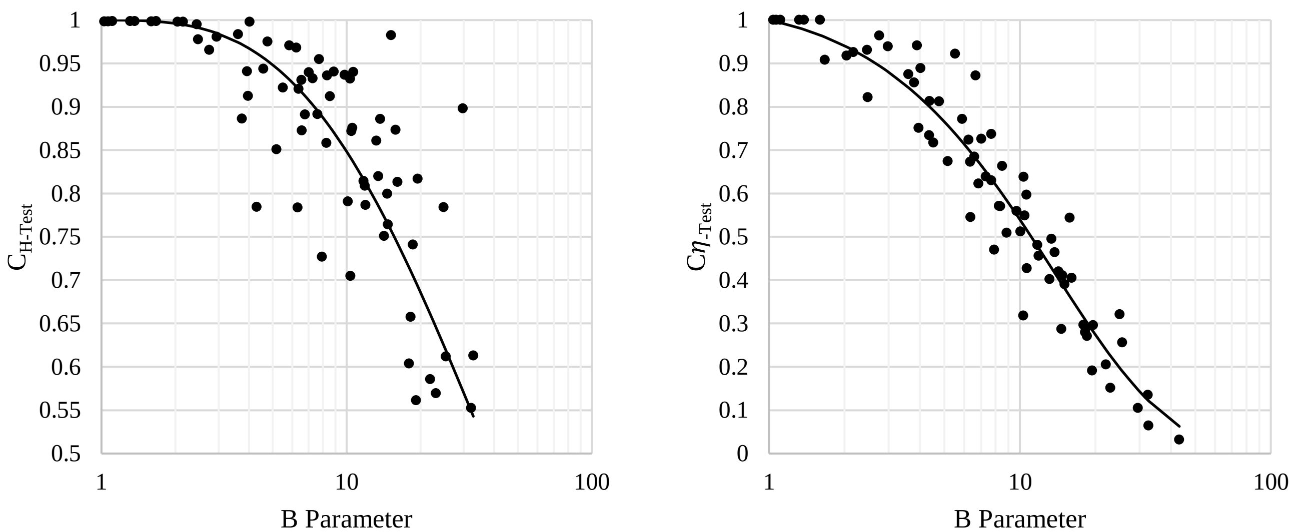

Figure 10. HI 9.6.7-2015 Basis Data (points) (Gülich 2000) with current model (solid line) indicated.

Before applying an analysis such as a prediction interval, it is critical that the variables are representing the intent of the analysis. If the statistical analysis is carried out as-is, they will show large uncertainty near correction values of one, and low uncertainty near values of zero. For this reason, a modification to the HI 9.6.7-2015 is used here for the purposes of statistical discussion. This modification is not intended to represent a suggested replacement of HI 9.6.7-2015, nor is it intended to represent a statistical analysis of full rigor.

The correction factors are representing how close the behavior is to that of water. The error of interest is not on the behavior near water, but on the viscous correction. Therefore, an indication of *correction strength* has been defined in Equation (10):

$$\mathrm{Z} = \sqrt{1 - C_{HI}} \tag{10}$$

The normal HI 9.6.7-2015 predictive model is still considered within this context – C_{HI} is any correction factor as defined by HI 9.6.7-2015. In this form, low values of B – near water behavior – will have narrow prediction intervals, whereas high values of B – more viscous situations – will have wide prediction intervals. This reflects the type of physical uncertainty expected.

The process for generating the prediction interval will not be presented here. For reference, see Navidi (2008) or other general statistics references. With the transformation above, 80% prediction intervals are shown in Figure 11 with dashed lines.

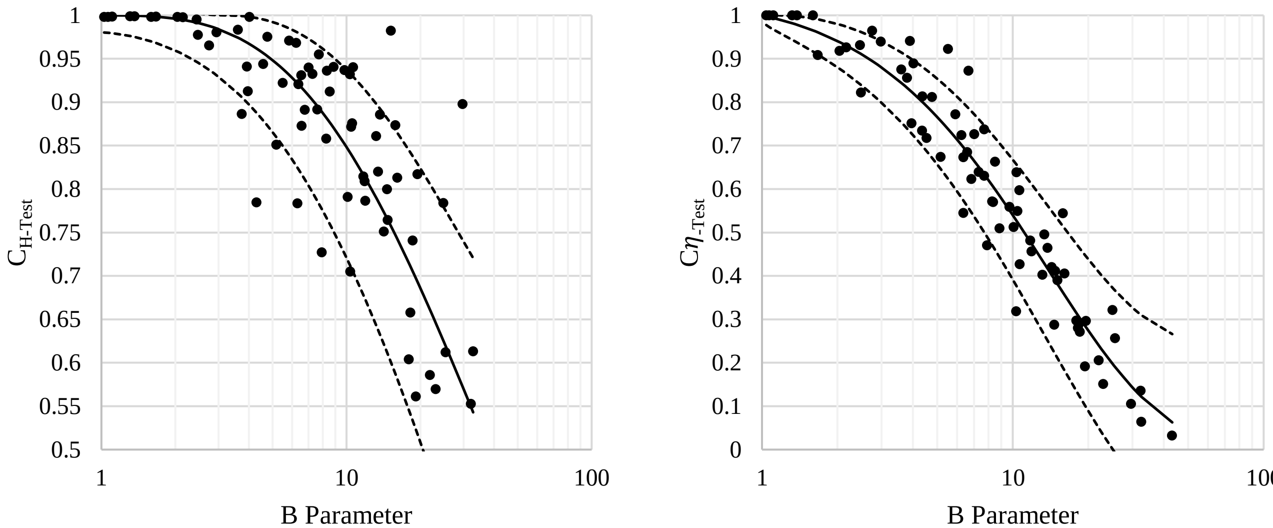

Figure 11. HI 9.6.7-2015 Data with 80% prediction intervals (dashed lines) based on transformation in Equation (10).

It is important to observe that, for a given B parameter, there are a range of possible correction factors. It should be expected that different pumps may have the same B parameter but somewhat different viscous performance. It is assumed here that this behavior follows a normal distribution. Any model based on a single dimensionless parameter will necessarily have this effect. An improved empirical fit to the data may be possible, but the uncertainty can never be driven to zero.

Comparison of Test Data and HI 9.6.7-2015 Basis Data

Whether or not the studies discussed here indicate concern for use of the HI 9.6.7-2015 guideline has yet to be addressed. To do so, the data is overlaid onto the HI 9.6.7-2015 Basis Data.

Additionally, the HI 9.6.7-2015 guideline currently contains an indication of uncertainty in the form of standard deviations on the correction factors. These values are 0.1 for the head correction factor, and 0.15 for the efficiency correction factor and Figures 12 and 13 indicate these bands. One standard deviation implies approximately 68% of pumps fall within the band $[C_{H-HI} - 0.1, C_{H-HI} + 0.1]$ for head, and similar for efficiency.

Figure 12 indicates estimated values of the head correction factor at BEP where possible. The solid black line “HI 9.6.7 Model” only predicts the head correction factor at BEP - the entire “New Test Data” series is shown only as a qualitative representation of how the head correction factors compare to the B parameter. Some test series do not have enough data to reliably estimate the viscous head at BEP, most notably the PRCI19 series, where only single test data points existed. For discussion on the outlying PRCI19.3 point, see the section Field Tests of Crude Oil Pipeline Pumps.

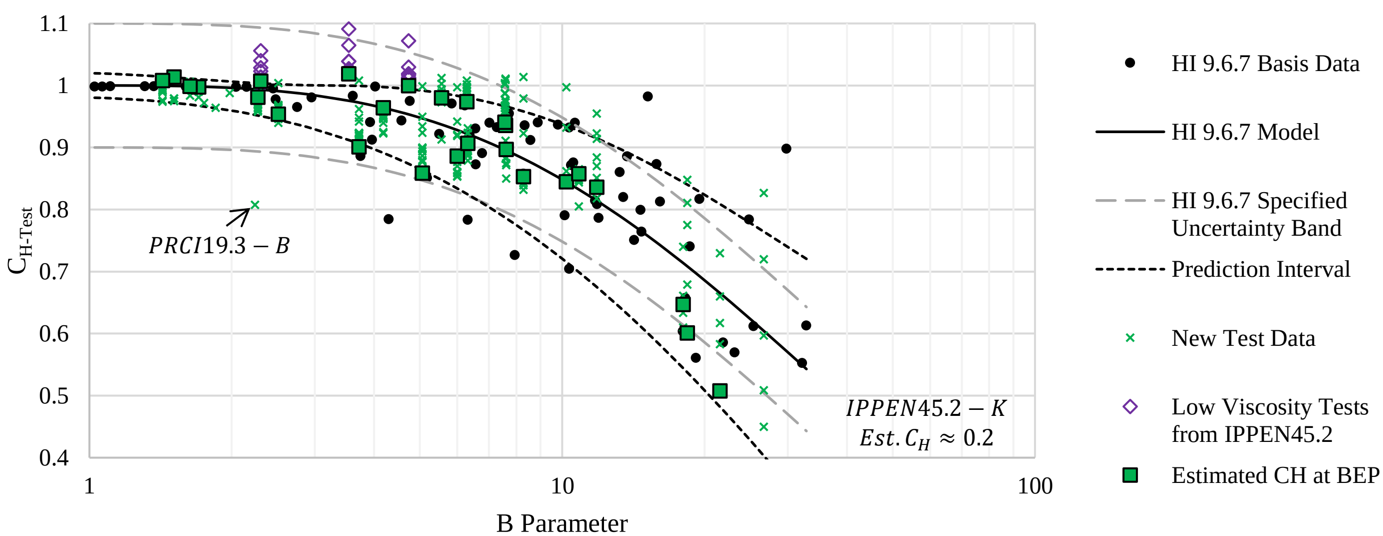

Figure 12. Test head correction factors from HI 9.6.7-2015 and recent tests, with uncertainty from HI 9.6.7-2015 and Prediction Interval.

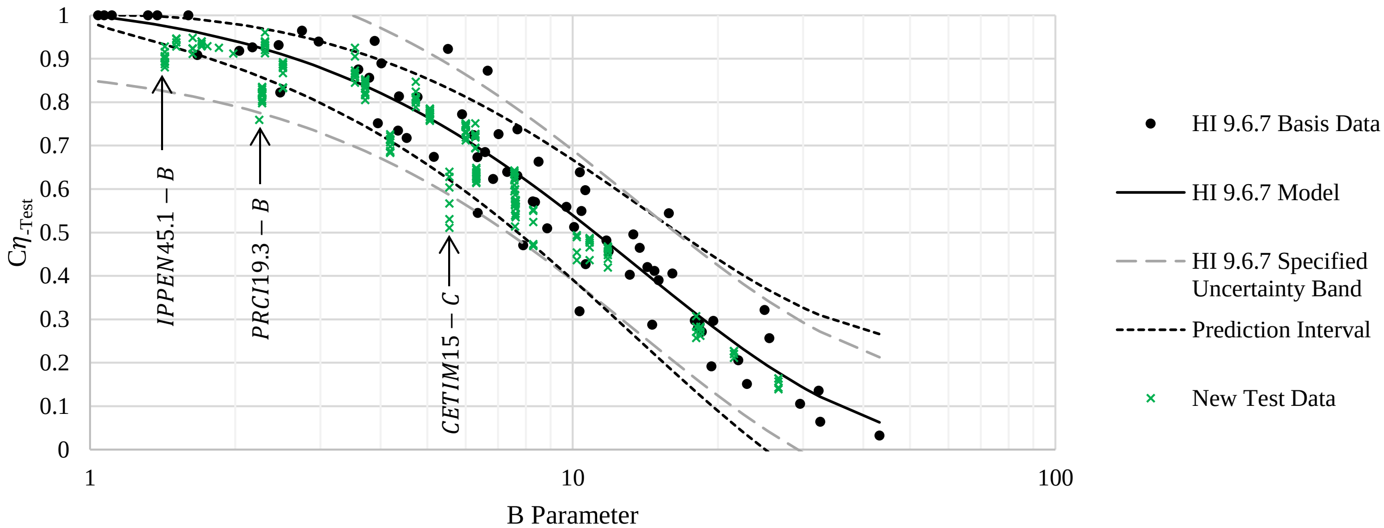

Figure 13. Test efficiency correction factors from HI 9.6.7-2015 and recent tests, with uncertainty from HI 9.6.7-2015 and Prediction Interval.

Most of the additional data falls within the 80% prediction interval, though there do appear to be trends for head and efficiency corrections. It appears that HI 9.6.7-2015 standard slightly overpredicts efficiency and slightly underpredicts head. Critically, however, this is data from only a handful of pumps. To "correct" the HI 9.6.7-2015 guideline with this data would heavily bias the prediction toward the five pump models considered. More data – from a wider variety of pumps and conditions – would be necessary to make any such adjustments.

Furthermore, the outliers occur mostly at extreme conditions, such as significant speed reduction. The head correction factors above one are likely due to increased volumetric efficiency before hydraulic or disk friction losses begin to increase substantially – an effect HI 9.6.7-2015 does not attempt to capture.

Applying Uncertainty in Pump and Motor Selection

In many typical engineering situations, the HI 9.6.7-2015 prediction will be precise enough as is, especially as various safety factors and margins of error are often used in any sizing process.

For sensitive or critical situations, a more conservative approach may be warranted. A conservative sizing should consider the uncertainty on the viscous prediction as described in this paper. It is recommended that the standard deviations as specified in HI 9.6.7-2015 be used to account for this uncertainty. After attaining a predicted correction factors from HI 9.6.7-2015, the head correction factor should be lowered by 0.1, and the efficiency correction factor should be lowered by 0.15. This would indicate the need for a potentially much larger pump and motor, but the confidence that the sizing is sufficient will be much higher.

In moderately sensitive applications, Figures 12 and 13 may be consulted for an uncertainty that depends on the parameter B. The uncertainty for head and efficiency specified in HI 9.6.7-2015 is likely overstated for low B values ($B < 10$), and perhaps understated for high B values ($B > 10$). This indicates that, for any application, more caution is warranted as the B value becomes larger and the predicted correction factors become stronger, or further from one.

Note that, for most applications, taking the above approach may not be necessary. At the time of this writing, neither the HI 9.6.7 Committee, nor several pump manufacturers or software companies implementing the corrections and involved with the Committee have received feedback that indicates HI 9.6.7-2015 is severely undersizing or oversizing equipment. It is possible that the uncertainties here are overstated, or that the error in prediction is overshadowed by other safety margins used during sizing.

Reducing Uncertainty

How can this uncertainty be reduced? As mentioned previously, any model that depends on a single dimensionless parameter (in this case, the B parameter) will by necessity represent a *distribution* of pumps with that value. No exactly certain calculation of viscous performance is possible without more fundamental changes to the model.

With the addition of a significant quantity of high-quality and varied data, the guideline could be improved. There are a few likely ways this could occur:

- The empirical constants may be adjusted to better represent the entire test data body.

- Additional data may allow the prediction interval to become smaller. Conversely, it could force it to become wider.
- Different estimations for corrections are developed for different types or classes of pumps.
- The guideline is expanded or amended to utilize more than just the B parameter in the estimation of the correction factors.

None of these are possible without additional data. The authors of this paper, and the HI 9.6.7 Committee, strongly solicit the industry to share with the Hydraulic Institute any relevant viscous test information toward this end.

Field Pump Performance Testing Procedure

Testing higher flow and power pumps with mineral oil based fluids in a laboratory is costly and poses environmental issues. Field pump performance testing is one solution, but extra care must be taken to achieve accurate test data. For this purpose, a report that provides practical methodology to accurately measure in situ pump viscous performance was developed by the PRCI (Robinett 2015). These in situ pump viscous performances along with the certified factory performance tests on water may be used to expand the database.

CONCLUSIONS

Any simple empirical model for predicting the performance of a centrifugal pump when the fluid viscosity is greater than that of water is subject to uncertainty. It is not reasonable to expect a perfectly accurate result from any such model. Quantifying the uncertainty of the model is as important as the accuracy of the model. The statistical analysis necessary is a complex field in its own right, and a simple analysis was presented here to indicate approximate uncertainty in the HI 9.6.7-2015 model.

The analysis here demonstrates that using a single value for uncertainty does not tell the whole story. In general, the uncertainty depends on the strength of the viscous correction. The HI 9.6.7-2015 method uses the dimensionless B Parameter to characterize the strength of the correction – higher viscosities, higher rotational speed pumps, or lower specific speed pumps all show larger viscous losses and have correspondingly larger B Parameters. Therefore, a low B Parameter (<5) has a low uncertainty (<10%), whereas a high B Parameter (>10) has a higher uncertainty (>20%). Users of any predictive viscous model are encouraged to understand this effect and provide margin in the sizing process as necessary.

Recent test data that may have shown concern over the HI 9.6.7-2015 predictions show results within these expected bounds. In fact, many of the tests agree quite well with the HI 9.6.7-2015 predictions, with less than 5% error in many cases. However, the current study certainly indicates the need for additional investigation into this topic.

With additional data, viscous performance models can be improved not only by more accurately predicting field behavior, but also by more accurately quantifying their uncertainty. Further data and analysis would allow the determination of more concrete uncertainties and sizing recommendations as a function of viscosity and pump design. Given enough varied data, the uncertainty could be made even lower by developing models specific to certain pump types or operating ranges.

No meaningful progress can be made toward any of these goals without more data, and pump manufacturers and users are called upon to aid in this endeavor.

NOMENCLATURE

$\eta_{Viscous}$ = Efficiency of a reduced curve, at a specific operating point
η_{Water} = Efficiency of a water curve, at a specific operating point
ω = Impeller rotational speed (rad/s)
B = HI 9.6.7-2015 B Parameter
BEP = Best Efficiency Point
C_η = Efficiency Correction Factor
$C_{\eta\text{-}Test}$ = Efficiency Correction Factor, as Tested
C_H = Head Correction Factor
$C_{H\text{-}Test}$ = Head Correction Factor, as Tested
C_{HI} = Generic Correction Factor, as predicted by HI 9.6.7-2015
C_Q = Flow Correction Factor
$C_{Q\text{-}HI}$ = Flow Correction Factor, as predicted by HI 9.6.7-2015
$H_{BEP\text{-}W}$ = Head of a water curve, at BEP
$H_{Viscous}$ = Head of a reduced curve, at a specific operating point
H_{Water} = Head of a water curve, at a specific operating point
N = Impeller rotational speed (rpm)

Nq = Specific Speed Metric Units, by ANSI/HI 1.1-1.2-2014 ($N_s = NQ_{BEP-W}^{0.5}/H_{BEP-W}^{0.75}$), using total flow through the pump for Q
N_S = Specific Speed US Units, by ANSI/HI 1.1-1.2-2014 ($N_s = NQ_{BEP-W}^{0.5}/H_{BEP-W}^{0.75}$), using total flow through the pump for Q
Q_{BEP-W} = Flow of a water curve, at BEP
$Q_{Viscous}$ = Flow of a reduced curve, at a specific operating point
Q_{Water} = Flow of a water curve, at a specific operating point
Re_B = Head Reynolds Number for a pump
Z = Correction strength

APPENDIX A – LIST OF CONSIDERED VISCOUS TESTS

Table 1 – List of viscous tests considered.

Test	Number of Test Points	Fluid	Kinematic Viscosity *cSt*	Specific Gravity	Shaft Speed *Rpm*	Water BEP Flow *gpm (m³/hr)*	Water BEP Head per stage *ft (m)*	B Parameter
CETIM15-A	11	Water	1	1.000	1783	5706 (1296)	192 (58.5)	1.00
CETIM15-B	9	Mineral Oil	1160	0.936	1783	5706 (1296)	192 (58.5)	7.59
CETIM15-C	6	Mineral Oil	518.7	0.936	1490	4768 (1083)	134 (40.9)	5.55
CETIM15-D	6	Mineral Oil	1155	0.936	1490	4768 (1083)	134 (40.9)	8.28
CETIM15-E	4	Mineral Oil	1751	0.936	1490	4768 (1083)	134 (40.9)	10.20
PRCI17-A	12	Water	1	1.000	1500	3610 (820)	123 (37.5)	1.00
PRCI17-B	10	Mineral Oil	87	0.868	1495	3597 (817)	122 (37.3)	2.51
PRCI17-C	12	Mineral Oil	191	0.877	1496	3602 (818)	122 (37.3)	3.72
PRCI17-D	12	Mineral Oil	354	0.884	1496	3602 (818)	122 (37.3)	5.06
PRCI17-E	12	Mineral Oil	498	0.887	1497	3602 (818)	123 (37.4)	6.00
PRCI19.1-A	6	Water	1	0.934	1795	18034 (4096)	758 (231)	1.00
PRCI19.1-B	4	Crude Oil	92	0.931	1795	18034 (4096)	758 (231)	1.51
PRCI19.1-C	4	Crude Oil	117	0.913	1796	18034 (4096)	758 (231)	1.70
PRCI19.2-A	6	Water	1	1.000	1795	16942 (3848)	719 (219)	1.00
PRCI19.2-B	4	Crude Oil	216	0.917	1792	16916 (3842)	715 (218)	1.63
PRCI19.3-A	6	Water	1	1.000	1795	16960 (3852)	728 (222)	1.00
PRCI19.3-B	1	Crude Oil	103	0.917	950	8978 (2039)	203 (62)	2.24
PRCI19.3-C	1	Crude Oil	103	0.917	1218	11509 (2614)	335 (102)	1.98
PRCI19.3-D	1	Crude Oil	103	0.917	1398	13209 (3000)	443 (135)	1.85
PRCI19.3-E	1	Crude Oil	103	0.917	1559	14732 (3346)	548 (167)	1.75
PRCI19.3-F	1	Crude Oil	103	0.917	1792	16933 (3846)	725 (221)	1.63
IPPEN45.1-A	24	Water	0.93	1	1895	1220 (277)	74 (22.5)	1.00
IPPEN45.1-B	11	Oil	15	0.9	1895	1220 (277)	74 (22.5)	1.43
IPPEN45.1-C	16	Oil	38	0.9	1895	1220 (277)	74 (22.5)	2.27
IPPEN45.1-D	12	Oil	129	0.9	1895	1220 (277)	74 (22.5)	4.19
IPPEN45.1-E	11	Oil	293	0.9	1895	1220 (277)	74 (22.5)	6.31
IPPEN45.1-F	9	Oil	427	0.9	1895	1220 (277)	74 (22.5)	7.62
IPPEN45.1-G	6	Oil	864	0.9	1895	1220 (277)	74 (22.5)	10.84
IPPEN45.2-A	11	Water	0.93	1	2875	260 (59)	136 (41)	1.00
IPPEN45.2-B	8	Oil	14	0.9	2875	260 (59)	136 (41)	2.30
IPPEN45.2-C	8	Oil	33	0.9	2875	260 (59)	136 (41)	3.54
IPPEN45.2-D	8	Oil	59	0.9	2875	260 (59)	136 (41)	4.73
IPPEN45.2-E	8	Oil	104	0.9	2875	260 (59)	136 (41)	6.28
IPPEN45.2-F	9	Oil	151	0.9	2875	260 (59)	136 (41)	7.57
IPPEN45.2-G	8	Oil	369	0.9	2875	260 (59)	136 (41)	11.83
IPPEN45.2-H	5	Oil	855	0.9	2875	260 (59)	136 (41)	18.01
IPPEN45.2-I	4	Oil	892	0.9	2875	260 (59)	136 (41)	18.40
IPPEN45.2-J	4	Oil	1226	0.9	2875	260 (59)	136 (41)	21.57
IPPEN45.2-K	5	Oil	1877	0.9	2875	260 (59)	136 (41)	26.69

*No specific gravity was given for individual IPPEN tests, the range of specific gravities for the oils tested was 0.87-0.93.

APPENDIX B – DERIVATION OF DIMENSIONLESS PARAMETER B

The parameter B is fundamental to the prediction method defined by HI 9.6.7-2015. Like Reynolds Number, or specific speed, this is a dimensionless parameter which can be derived from the homologous pump laws.

Centrifugal pump performance may depend on; flow Q, pressure rise Δp, power P, diameter D, speed ω, fluid density ρ, and fluid viscosity μ. These 7 functional variables have 3 dimensions (M, L, T), and from the Buckingham Pi Theorem therefore describe 4 dimensionless groups. Choosing D, ω, and ρ as the common variables, we can determine the following dimensionless groups.

$$\Pi_1 = \frac{Q}{D^3\omega} \tag{11}$$

$$\Pi_2 = \frac{D^2\omega^2\rho}{\Delta p} = \frac{D^2\omega^2}{g\Delta H} \tag{12}$$

$$\Pi_3 = \frac{P}{D^5\omega^3\rho} \tag{13}$$

$$\Pi_4 = \frac{\mu}{D^2\omega\rho} = \frac{1}{Re} \tag{14}$$

It is useful to consider the definition of specific speed, which is a combination of Π_1 and Π_2 such that diameter D is cancelled and ω has an exponent of 1.

$$N_S^* = (\Pi_1^2\Pi_2^3)^{1/4} = \left[\left(\frac{Q^2}{D^6\omega^2}\right)\left(\frac{D^6\omega^6}{g^3\Delta H^3}\right)\right]^{1/4} = \frac{Q^{1/2}\omega}{g^{3/4}\Delta H^{3/4}} \tag{15}$$

Equation (15) is the true dimensionless specific speed. In practice, the gravitational term in the denominator is usually omitted, imparting units of dimension $\left[\frac{L}{T^2}\right]^{3/4}$ onto the quantity.

The parameter B is formed from a different set of dimensionless groups.

$$B^* = \left(\frac{\Pi_4^8}{\Pi_1^6\Pi_2}\right)^{1/16} = \left[\left(\frac{\mu^8}{D^{16}\omega^8\rho^8}\right)\left(\frac{D^{18}\omega^6}{Q^6}\right)\left(\frac{g\Delta H}{D^2\omega^2}\right)\right]^{1/16} = \frac{\nu^{1/2}g^{1/16}\Delta H^{1/16}}{Q^{3/8}\omega^{1/4}} \tag{16}$$

This equation is similar in form to an inverse specific speed, but with the addition of kinematic viscosity, and differing exponents. It is possible with this definition to have two pumps with identical specific speeds and identical Reynolds Numbers as defined by Equation (14), but different B parameters.

However, it is also possible to define a Reynolds Number with flow and head, as opposed to the more general definition above.

$$\frac{1}{Re_B} = \frac{\nu}{Q^{1/2}g^{1/4}H^{1/4}} \tag{17}$$

This version of Reynolds Number can be justified by stating that head is proportional to velocity squared, flow area is proportional to diameter squared, and velocity is proportional to speed times diameter

$$H \propto V^2 \rightarrow V^{1/2} \propto H^{1/4} \tag{18}$$

$$A \propto D^2 \rightarrow VA = Q \propto VD^2 \rightarrow V^{1/2}D \propto Q^{1/2} \tag{19}$$

$$V \propto D\omega \tag{20}$$

Combining Equations (18 – 20) with the denominator of Equation (14)

$$D^2\omega \propto VD = V^{1/2}V^{1/2}D \propto H^{1/4}Q^{1/2} \tag{21}$$

Which implies

$$\frac{1}{\Pi_4} = \frac{1}{Re} \propto \frac{1}{Re_B} \tag{22}$$

Taking this definition, we can see that the B Parameter is directly related to Reynolds Number and specific speed.

$$B \propto \frac{1}{Re_B{}^{1/2}Ns^{1/4}} \tag{23}$$

This definition was used by Gülich in the development of the HI 9.6.7-2015 Basis Data (Gülich 2000) and is implied by the nomograms in ANSI/HI 1.1-1.5-1994.

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