Preliminary Hydraulic Design and Test of A Centrifugal Blood Pump: Effects of Reynolds Number and Blade Number

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Abstract

In our design and model test of a centrifugal blood pump as described in Anansukkasam *et al.* [1], two questions arise. The first question is regarding the effect of Reynolds number on the dimensionless head-flow ($C_H - C_Q$) curve of the pump. The second question is regarding the suitable number of blades for the impeller. Consequently, this paper has the objectives in investigating the effects of Reynolds number and blade number on the $C_H - C_Q$ curve of the designed pump. The experiment is conducted over a wide span of Reynolds number, a factor of 100, from 0.17 to 18.6 times of the predicted prototype Reynolds number (Re_p) and with 4- and 6-blade impellers. The results show that the $C_H - C_Q$ curve changes relatively little over what can be considered the high Reynolds number range, approximately from 18.6 to 1 Re_p, a span of a factor of 20. However, C_H does decrease more significantly as the Reynolds number decreases into the lower Reynolds number range, approximately from 1 to 0.17 Re_p, a span of only a factor of 5. In addition, the effect is more pronounced towards the high C_Q end than the low C_Q end. As for the effect of blade number, the results show that the 6-blade impeller gives higher C_H than the 4-blade impeller, both in the high and low Reynolds number ranges, but being more pronounced in the high Reynolds number range. Finally, the prediction of the prototype speed in [1] is evaluated. It is found that at the prototype-equivalent Reynolds number, which can be considered at the end of the high Reynolds number range, the pump delivers the head slightly lower than predicted, by approximately 10%, albeit still within the uncertainty. On the one hand, this can be attributed to the effect of Reynolds number since in [1] the much higher Reynolds number $C_H - C_Q$ correlation is employed in the prediction. On the other hand, the fact that the predicted prototype Reynolds number is at towards the end of the high Reynolds number range results in only slight deviation from the prediction. Finally, qualitatively, the implications of the effects of Reynolds number are 1) the use of (Reynolds number independence) similarity law for scaling the $C_H - C_Q$ curve is reasonably valid at high Reynolds number range, but Reynolds number effect may need to be taken into account at low Reynolds number range, *especially* towards the high C_Q end, and 2) as far as the $C_H - C_Q$ curve is concerned, the choice of the suitable number of blades is more critical at high Reynolds number than at low Reynolds number.

Keywords: centrifugal blood pump, head-flow, similarity law, Reynolds number, blade number

1. Introduction

In our design of a centrifugal blood pump [1], two questions concerning the effects of Reynolds number and blade number arise. For the Reynolds number, the question of the effect of Reynolds number on the dimensionless head-flow curve arises in the context of the similarity and the prediction of the operating speed of the prototype from the model testing data, given that the prediction in that work is conditioned by the assumption of Reynolds number independence of the dimensionless head-flow curve. More generally, though, the question of the effect of Reynolds number on the dimensionless head-flow curve can peripherally help selecting suitable design condition. For the blade number, the natural question in the design process is regarding the suitable number of blades for the desired dimensionless head-flow. For further details, see [1].

In this regard, Day *et al*. [2] studied the effects of Reynolds number on the effectiveness of the traditional pump affinity laws and on the performance of a small centrifugal blood pump. Their pump is backwardly-curved 5-blade pump with impeller diameter of 46 mm. They found that, for their pump, while higher Reynolds number flows scale effectively according to the conventional affinity laws, lower Reynolds number flows do not adhere to the laws. As a result, it can be concluded that the consideration of Reynolds number is necessary for the scaling and design of the pump. In addition, in their work they varied Reynolds number by a factor of 30 and found that the head coefficient at the design flow coefficient varied by a factor of approximately 1.5. It is important to note here that in their work they defined the dimensionless numbers differently from the present work, especially the Reynolds number which they defined it based on the volume flowrate rather than the rotational speed like in the present work.

Rababa [3] studied the effect of blade number and showed that the head-flow and efficiency increased with blade number. Liu *et al*. [4] found similar result for the effect of blade number on the head. Li *et al.* [5] reported that the number of blades has a strong influence on the performance of centrifugal oil pumps, but as the viscosity of liquid pumped increases this influence weakens.

While past works have suggested to an extent the effect of Reynolds and blade numbers, naturally in the design work such as described in [1], we are interested in more specifically the influences of both that are specific to our design. In this regard, the present work has the following objectives: 1) the investigations of the effects of Reynolds number and blade number on the dimensionless head-flow $(C_H - C_Q)$ curve of the designed centrifugal blood pump in [1] and the validity of the assumed Reynolds number independence of the $C_H - C_Q$ curve given in [1] down to the predicted prototype Reynolds number; and 2) the evaluation of the prediction of the operating speed of the prototype (1,185 RPM) as described in [1].

2. The Centrifugal Blood Pump

The design and configuration of the centrifugal blood pump used in this experiment is described in [1]. Briefly, Table 1 summarizes the geometric parameters of the *prototype* impeller, while Fig. 1 shows the 2X scale-up *model* of the pump with 6-blade impeller installed. For further details, see [1].

Table 1. The geometry of the *prototype* impeller.

Geometric Variables				
β_1 (deg)	7.73°		β_2 (deg)	78.57°
d_1 (mm)	13.16		d_2 (mm)	65.80
b_1 (mm)	25		b_2 (mm)	5
λ_d	0.2		λ_b	5
θ (deg)	70.84		-	-

Notation

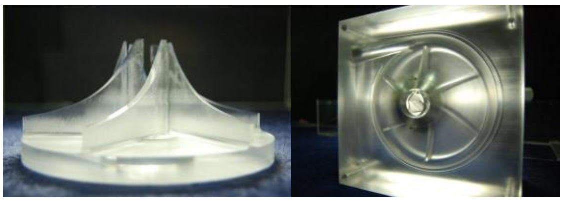
1 = impeller inlet, 2 = impeller exit

β = blade angle, θ = blade subtended angle

d = impeller diameter, b = impeller width

$\lambda_d := d_1 / d_2$, $\lambda_b := b_1 / b_2$

For further details, see [1].



(a) (b)

Fig. 1. The 2X scale-up model: (a) impeller, (b) assembly of the impeller and the pump casing.

3. Similarity Scaling and Model Testing

In the evaluation of the effects of Reynolds number (Re) and the number of blades (n) on the dimensionless head-flow curve ($C_H - C_Q$ curve), we employ the similarity scaling law

$$C_H = f\left(C_Q; \mathrm{Re}, n; B\right), \tag{1}$$

where C_H is the dimensionless head coefficient, $C_H := gH / \omega^2 d^2$, C_Q is the dimensionless flow coefficient, $C_Q := Q / \omega d^3$, Re is the Reynolds number, $\mathrm{Re} := \omega d^2 / \nu$, n is the blade number, and B is the blade blockage ratio at the impeller inlet. Here, H is the total hydraulic head; ω angular velocity of the impeller; Q volume flowrate; and ν kinematic viscosity of fluid. In addition, for the characteristic diameter d, we choose the impeller exit diameter d_2.

In writing the functional form in Eq. (1), we use the semicolonned three-slot convention. That is, the roles of variables in the parentheses are classified and separated by two semicolons according to (independent variables ; variable parameters ; constant parameters), where in this case the independent variable is C_Q; the variable parameters are Re and n, the parameters we want to vary in the experiment and investigate their effects; and the constant parameter is B, the parameters that are kept fixed throughout the experiment.

Range of The Parameter Space

Finally, according to the functional form, Eq. (1), in the present experiment we vary Reynolds number Re over the range of $2.4\text{x}10^4$ to $2.6\text{x}10^6$, or 0.17 to 18.6 times of the predicted prototype Reynolds number, a variation by a factor of approximately 100. In addition, we experiment with two numbers of blades n, 4 and 6. For further details of each case, see Table 2 in Sec. 4. Here, the blade blockage ratio B is defined as the total radial area blocked to the total radial area at the impeller inlet, $B = nt\sin\beta_1 / (\pi d_1)$, where t is the blade thickness. For the two numbers of blades, 4 and 6, the blockage ratios are 2.93% and 2.60%, respectively. While they are not exactly equal, they differ by only 12% with respect to the

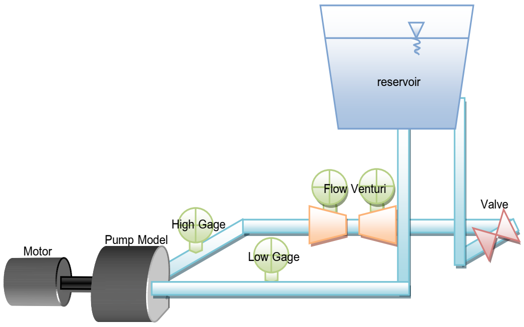


Fig. 2. Schematic diagram of the pump test rig.

minimum (2.6%) and only 0.33% with respect to the total impeller inlet area. As a result, they are deemed practically equal, and B is considered a constant parameter over the whole experiment.

4. Experimental Setup and Conditions

Experimental Setup

The setup for the model testing is the same as in [1] and is shown diagrammatically in Fig. 2. Note that the model pump is a 2X scale-up of the prototype. Thus, the dimensional parameters of the model are twice the values for the corresponding parameters of the prototype tabulated in Table 1. Briefly, the setup is a closed loop test rig with upper reservoir open to atmosphere. Two pressure gauges are installed at 20 pipe diameters away from the pump inlet and exit for the measurement of static pressures. Volume flowrate is measured by a venturi. The total hydraulic head is determined from

$$H = \left(\frac{p}{\rho g} + \frac{\alpha \bar{V}^2}{2g} + z\right)_2 - \left(\frac{p}{\rho g} + \frac{\alpha \bar{V}^2}{2g} + z\right)_1, \quad (2)$$

where, because of equal inlet and exit pipe diameters, the total hydraulic head depends essentially on the static pressure rise across the pump. Also, note that the reading of the pressure gages is corrected for the elevation.

In order to vary Reynolds number over the wide range, we use water and glycerin/water solutions at various mixture ratios. In order to vary the blade number, we make one volute and casing but two impellers of the same geometric parameters, one with 4 and another with 6 blades. In order to vary the number of blades, we swap the two impellers in and out of the same volute and casing. Therefore, for the test of both impellers, the volute and casing is one and the same.

Experimental Conditions

All cases runs are tabulated in Table 2, together with their designation and condition for ease of reference. In this table, we use the Reynolds number of the predicted prototype (Re_p =1.4x10^5) as a reference, and the cases 1 to 6 have the Reynolds number of 18.6, 9.3, 3.4, 1.0, 0.86, and 0.17 times of Re_p, respectively. Note that the predicted prototype Reynolds number refers to the Reynolds number of the predicted prototype (case 4) after incorporating the model test data in [1], which was predicted to operate successfully at 1,185 RPM. On the other hand, the originally design Reynolds number (case 5) is the Reynolds number at the originally designed RPM of 1,000. The range of tested Reynolds number therefore spans a factor of 100 (or 110 to be more exact). Cases a and b designate the cases of 4-blade and 6-blade impellers, respectively.

Table 2. Case designation and condition.

Case[(1)]	Fluid[(2)]	ν (m^2/s)	N (RPM)	Re_m	$\mathrm{Re}_m/\mathrm{Re}_p$
1a,1b	water	8.03x10^{-7}	1,185	2.6x10^{6}	18.6
2a,2b	water	8.03x10^{-7}	600	1.3x10^{6}	9.3
3a,3b	50% glycerin	3.76x10^{-6}	1,000	4.8x10^{5}	3.4
4a,4b	72.7% glycerin	1.52x10^{-5}	1,185	1.4x10^{5} (3)	1.0
5a,5b	72.7% glycerin	1.52x10^{-5}	1,000	1.2x10^{5} (4)	0.86
6a,6b	89% glycerin	7.62x10^{-5}	1,000	2.4x10^{4}	0.17

(1) Case a = 4-blade impeller, b = 6-blade impeller

(2) per cent is per cent of the total weight of the solution, e.g., 72.7% glycerin means the ratio of glycerin weight to the total weight of the solution is 0.727.

(3) predicted prototype Reynolds number (predicted at 1,185 RPM – after incorporating model testing data in [1])

(4) originally design Reynolds number from Euler's equation (originally designed at 1,000 RPM)

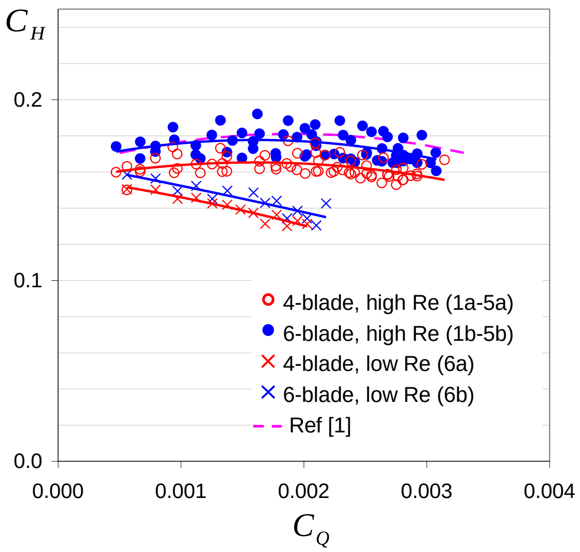


Fig. 3. Effects of Reynolds number and blade number at high and low Reynolds number.

5. Results and Discussions

5.1. Effects of Reynolds Number and Blade Number

Effects of Reynolds Number

Figure 3 shows the result for the dimensionless head-flow coefficients ($C_H - C_Q$) for all cases experimented. For each blade number, the results are presented in two groups: 1) high Reynolds number range, which includes case 1 to 5 where Re = 0.86-18.6 Re_p, and 2) low Reynolds number range, which includes case 6 where Re = 0.17 Re_p. The uncertainties of both C_Q and C_H are estimated to be $\pm$ 10% of the reading value. Firstly, because of the uncertainty in measurement, we feel that our results are not warranted to be differentiated among the first five cases, hence the groupings.

As for the effect of Reynolds number, for both cases of 4- and 6-blade impellers, the results show that in the high Reynolds number range C_H changes relatively little (within the limit of the uncertainty) over the decrease in Reynolds number by a factor of 20, from 18.6 to 0.86 Re_p. However, in the low Reynolds number range (case 6), C_H decreases more significantly, especially towards the high C_Q end than the low C_Q end, over the decrease in Reynolds number by a factor of only 5 (from case 5 to 6).

Qualitatively, the implication of this result on the effect of Reynolds number on the use of Reynolds number independence similarity, or affinity, law for scaling the $C_H - C_Q$ curve of the pump is then the following. At high Reynolds number, the Reynolds number independence scaling is reasonably valid, and the similarity law of the form

$$C_H = f(C_Q; \cdot ;...) \quad (3)$$

can be used, given that all other things being equal except the Reynolds number, which need not be. However, at low Reynolds number, care must be taken in using the similarity law, and the effect of Reynolds number may need to be taken into account by using the similarity law of the form

$$C_H = f(C_Q; \mathrm{Re}; ...), \quad (4)$$

especially at high C_Q end.

Quantitatively, though, the transition from high to low Reynolds number may depend on other factors such as the specific design of the pump. In this regard, the present result is qualitatively consistent with the result of [2]. Bear in mind however that the two works use different definition of Reynolds number.

Finally, the best fit curves at high Reynolds number for the present work can be given as

4-blade [Re = 0.86-18.6 Re_p]:

$$C_H = -4{,}033 C_Q^2 + 12.89 C_Q + 0.1549 \quad (5a)$$

6-blade [Re = 0.86-18.6 Re_p]:

$$C_H = -4{,}975 C_Q^2 + 16.03 C_Q + 0.1647. \quad (5b)$$

When compared to the correlation given in [1] (also shown in Fig. 3), whose result is for the same 6-blade impeller but over the higher and narrower range of Reynolds number,

6-blade [Re = 10-20 Re_p]:

$$C_H = -5{,}423 C_Q^2 + 20.66 C_Q + 0.1615, \quad (6)$$

we see that the present curve gives lower C_H than that given in [1] by approximately 5% towards the high C_Q end. This is consistent with the above result on the effect of Reynolds number since the correlation given in [1] is from the higher range of Reynolds number.

As for the assumption made in [1] regarding Reynolds number independence of Eq. (6) for the prediction at the prototype Reynolds number, we see that Eq. (6) is reasonably valid and can be assumed Reynolds number independence down to the prototype Reynolds number if a possible over-estimate error in the order of 10% is acceptable. Otherwise, Eq. (5b) can be suggested instead.

Effects of Blade Number

As for the effect of blade number, from Fig. 3 we see that, as far as the $C_H - C_Q$ curve is concerned, the 6-blade impeller gives relatively higher C_H than the 4-blade impeller, both at high and low Reynolds number ranges. The result also indicates that the effect of blade number is more pronounced at high Reynolds number than at low Reynolds number. These results are qualitatively consistent with what is reported in [5].

Qualitatively, the implication of this result in terms of the design of the pump is then the following. The choice of the suitable number of blades is more critical at high Reynolds number than at low Reynolds number.

5.2. Evaluation of The Prediction of The Operating Speed in [1]

To recap, in [1] it is concluded that, if the dimensionless head-flow curve of this pump, Eq.

(6), is independent of Reynolds number down to the prototype Reynolds number, the prototype should be able to deliver the desired head-flow successfully at 1,185 RPM. The remaining issue raised there is whether the assumption of Re independence down to the prototype Re is valid.

We have already addressed the issues of the effect of Reynolds number and the Reynolds number independence of the $C_H - C_Q$ curve in Sec. 5.1. Finally, in order to evaluate the prediction of the operating speed as described in [1] more clearly, we plot the dimensional result of case 4b ($\mathrm{Re} = 1.4 \times 10^5$) in Fig. 4. Also shown in the figure for comparison are the ideal $H - Q$ line at the originally design 1,000 RPM, and the former test data – using water - at the originally design 1,000 RPM ($\mathrm{Re} = 2.3 \times 10^6$) and at the predicted 1,185 RPM ($\mathrm{Re} = 2.7 \times 10^6$), both are from [1], which is at much higher Reynolds number than case 4b.

To recap, in [1] we originally design the pump at 1,000 RPM using the ideal Euler's equation (neglect all losses), test it at 1,000 RPM, and find that the head is only approximately 70% of the ideal value (pink rectangle markers in Fig. 4). We then make more tests, spanning the high Reynolds number range of approximately 10 to 20 Re_p, and find that the $C_H - C_Q$ curve, Eq. (6), is basically independent of Reynolds number in this high Re range. We then assume that the curve is Reynolds number independence and use similarity law to predict the operating point of the prototype to be 1,185 RPM. We later test the prediction by doing another model test at 1,185 RPM (green circle markers in Fig. 4), using

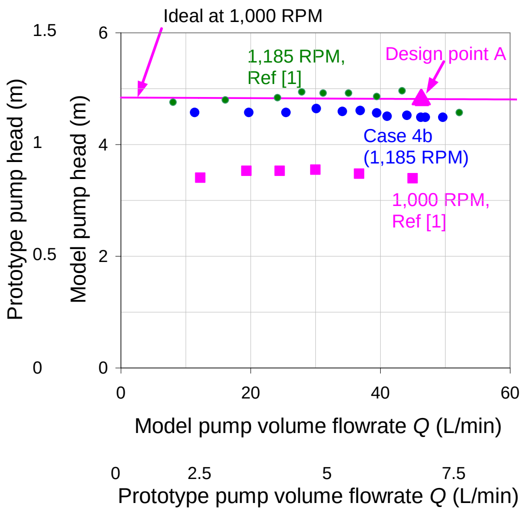


Fig. 4. Evaluation of the prediction in [1] at 1,185 RPM.

water which results in $\mathrm{Re} = 2.7 \times 10^6$ - much higher than the prototype Reynolds, and find that the model achieves the prototype-equivalent operating head-flow. Yet the question remains whether we can achieve the desired head-flow when we actually run the prototype at the prototype Reynolds number, $\mathrm{Re}_p = 1.4 \times 10^5$.

The result for case 4b, which is the prototype-equivalent case, shown as blue circle markers in Fig. 4 shows that the effect of Reynolds number at this relatively high Reynolds number range ($\sim \mathrm{Re}_p$ up, see Sec. 5.1) causes slight lowering in the head-flow curve as Reynolds number decreases from 2.7×10^6 to Re_p, resulting in slightly lower head. The operating of the model at the prototype-equivalent Reynolds number then delivers less head than the desired and predicted value at the design point A, albeit still within the uncertainty.

6. Conclusions

The effects of Reynolds number and blade number on the performance of the centrifugal blood pump designed and reported in [1] are investigated. The range of Reynolds number investigated is from a factor of 20 higher than the predicted prototype Reynolds number (Re_p) to a factor of 5 lower than Re_p, the total span of a factor of 100. The results show that in the high Reynolds number range, approximately from Re_p up, Reynolds number has relatively little influence (within $\pm$ 10%) on the $C_H - C_Q$ curve. However, in the low Reynolds number range, approximately from Re_p down, the effect of Reynolds number becomes more pronounced, causing the reduction in C_H as Reynolds number decreases, *especially* towards the high C_Q end than the low C_Q end. As for the effect of blade number, the results show that the 6-blade impeller gives higher C_H than the 4-blade impeller, both at high and low Reynolds number ranges. The result also indicates that the effect of blade number is more pronounced at high Reynolds number range than at low Reynolds number range.

The evaluation of the prediction of the operating speed of the prototype as described in [1] is simulated here by operating the model at the prototype-equivalent Reynolds number. The results show that, because of slight Reynolds number effect, the prototype-equivalent model (case 4b) renders slightly lower head than the predicted head, albeit still within the uncertainty.

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