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The effect of balancing holes on performance of a centrifugal pump: numerical and experimental investigations

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Abstract. In the present work, the effect of balancing hole's diameter on the performance of a centrifugal pump is experimentally and numerically investigated. Normally, in centrifugal pumps, design of impeller and volute is carried out in such a way that pressure distribution on both sides of the impeller becomes similar, thus minimizing the axial force exerted on the pump impeller and bearings. However, in reality this goal is not totally fulfilled and due to design and manufacturing circumstances and limitations, significant axial forces are generated. As a result excessive load on bearings, noise, vibration and damage to pump performance are experienced. In order to reduce the axial force, several methods are proposed. The simplest method is drilling balancing holes on the impeller eye. In this study, numerical and experimental investigations have been conducted to study the effect of balancing holes on a centrifugal pump performance and axial force reduction. A test rig is designed and equipped with measuring instruments. Experimental data for pump performance including flow, head, power and efficiency were obtained. Next, numerical results were validated against experimental data and a good agreement was observed. It was found that increasing balancing holes diameter up to 5 mm can lead to 5.6% reduction in head at design point. Furthermore, it was found that the diameter of balancing hole affects the pump efficiency and can reduce it by 2.6%. Numerical investigation showed that the diameter of balancing hole can play a prominent role on the axial force reducing it significantly. Extreme care must be taken in selecting the diameter of balancing holes since increasing holes diameter beyond a specific value does not reduce the axial force significantly, but, still aggravates the pump performance undesirably. Therefore, an optimum diameter of 3.5 mm was selected whereby the axial force is reduced up 56% at design point while reduction in performance characteristics are acceptable.

1. Introduction

In centrifugal pumps, bearings are exposed to radial and axial forces. Axial force is due to flow pattern in the impeller. In reality, due to design circumstances and limitations, the pressure distribution in front and rear of the impeller are not the same and as a result, a force in the direction of rotation axis of impeller is generated. Balancing and controlling this force is highly important, because it can make damages to parts of the pump and therefore pump life is reduced [1].

Nomenclature:

Symbol	Unit	Expression	Description
d	mm		Blade outer diameter
F_a	N		Axial force
F_a^*	-	$F_a/(\rho u_2^2 d^2)$	Coefficient of Axial force
g	m/s^2		Standard gravity acceleration
H, H_n	m		Head and design head
N_s	[r/min, m^3/s, m]	$NQ^{0.5}H^{-0.75}$	Metric specific speed
N	r/min		Rotational speed
Q, Q_n, Q_l	m^3/s		Flowrate, design flowrate and leak flowrate
Re	-	$(\Omega {d_2}^2)/\upsilon$	Pump Reynolds number
W	Watt		Power
Z			Number of blades

Greek Letters

Symbol	Unit	Equation	Description
η	-	$\rho g Q H\ /(W)$	Pump hydraulic efficiency
υ	m^2/s		Kinematic viscosity
ρ	kg/m^3		Fluid density
$\delta, \delta_n, \delta_l$	-	$Q/(\Omega d^2)$	Flow coefficient, design flow coefficient and leakage flow coefficient
ε	-	$\delta_l/\delta_n \times 100$	Flow coefficient and design flow coefficient
ψ, ψ_n	-	$gH/(\Omega^2 d^3)$	Head coefficient and design head coefficient
Π	-	$P/(\rho \Omega^3 d^5)$	Power coefficient
Ω	Rad/s		Angular velocity

One of the most challenging areas in the design of pump shaft is calculating the axial force exerted on it. This force is affected by the pump performance characteristics, operating status and internal pressure fields. One of the earliest studies on the evaluation of this force backs to the research accomplished by Lobanf and Rose [2]. They proposed a set of correlations for predicting the axial force in open, semi-open and closed impellers for one stage pumps. These correlation are somewhat reliable only for one stage pump, but complexities in the system design specifically multi-stage pump leads to a difference between the proposed results and experiments.

Several methods have been developed to reduce the axial forces according to the impeller size and geometry, number of stages, manufacturing process and the working fluid [3]. One of the popular methods in multi-stage pumps is using balance piston and balance disk. Salvadori et al. [4] numerically simulated fluid flow in a multi-stage pump and studied the effect of each stage on the axial force by considering the internal flows and pressure distributions. Additionally, they studied the effect of leakage flow and shroud chamber on the axial force. In a similar research, Salvadori et al. [5] studied a multi-stage pump with piston balance and their result showed better agreement with experiments at lower flowrates. It is worthwhile to be mentioned that using long balance drum are not preferable as they can lead to rotor dynamic instabilities in the system [6].

Similar studies have been also carried out on the calculation of axial force and side chamber effects like those conducted by Gantar [7] and Gata et al. [8]. In the research conducted by Gantar, the pressure and velocity distribution inside chamber were studied numerically and experimentally. Gata et al. used 2-D and 3-D simulations of fluid flow inside the side chamber for obtaining the axial force.

They reported that the side chamber has a remarkable effect on the axial force and its pressure distribution is affected by the leakage flow and local rotational speed inside the chamber.

Cutting radial grooves which is called "J-groove" on the front case of the pump is another practice utilized to reduce the axial thrust. This method can beneficially improve the stability of pump performance characteristics [9]. Another methods which is capable of canceling the axial thrust is introducing radial ribs so called as "back vanes" to the shroud of the impeller [10]. This method promotes swirl within the rear chamber and as a consequence the static pressure is dropped within the chamber. Godbole [11] experimentally made an investigation on the effect of back vanes and found 4% efficiency drop in the pump at Best Efficiency Point (BEP). Cao et al. [12] numerically studied the effect of back vanes and they reported that back vanes can reduce internal leakages. Similar numerical study was performed by Mortazavi et al. [13] in which the effects of design parameters of back vanes such as outer radius, width, clearance, and vane angle on pump performance and axial thrust were investigated.

Although a number of researches have been dedicated to study the axial force, no specific study has been concerned on the effect of balancing holes on the radial force and performance characteristics as head, power and efficiency. In the current study, it is aimed to study the effect of diameter of balancing holes of impeller on the axial force as well as characteristic parameters numerically and experimentally. In this work, the fluid flow is first simulated and will be verified against experiments. Thereafter, the axial force is obtained from numerical results and the effect of balancing hole diameter will be investigated. It is expected that in the current the research the optimum hole size is achieved so that the axial force is reduced and hydraulic losses are within accepted range.

2. Experimental setup

Sa In the present study, a centrifugal pump with a specific speed of $\mathrm{N_s} = \mathrm{NQ^{0.5}H^{-0.75}} = 57$ and a Reynolds number of $\mathrm{Re} = (\Omega d^2)/\upsilon$=1.56×10^6 is investigated. As specified in **Table 1** the pump impeller has 5 blades and its head and flow coefficients at design point are ψ_n =0.064 and δ_n =0.019 respectively. The outer impeller of pump is 65 mm.

In order to conduct experimental analysis a test rig was manufactured. The test setup has a reservoir and the pump is submerged within the tank. In the pump discharge line a control valve is installed to adjust the outlet pressure of the pump. The outlet pressure is measured and indicated using a pressure transducer. The flow is then passed through a venturi flowmeter and by measuring the pressure difference in two sides of the venturi, the flowrate is measured. The rotational speed of the pump is recorded by a magnetic rotor meter and the input power is measured as well. Using measured head, power and efficiency the performance curves of the pump are obtained.

The accuracy of instruments used in the present study is given in **Table 2**. Considering errors given in **Table 2**, uncertainties of flowrate, head, power, and efficiency are found to be 0.5%, 1%, 0.5%, and 2% respectively.

Table 1.pump characteristics

Parameter	Value
Specifics speed, N_s	57
Reynolds number, *Re*	1.56×10^6
Nominal discharge coefficient, ϕ_n	0.019
Nominal head coefficient, ψ_n	0.064
Impeller outer diameter (mm)	65
Blades Number, Z	5

Table 2. Measuring instrument accuracy

Instrument	Unit	Accuracy
Manometer	*m*	0.001
Outlet pressure transducer	*bar*	0.001
Power	*bar*	1
Rotor meter	*rpm*	1

3. Drilling balancing holes

Each performance test on the pump was carried out with an impeller with a specific balancing hole diameter. In the first test, the impeller has no balancing hole (See Figure 1) and the performance characteristics are measured. In the next steps, at each step, by drilling the impeller, the diameter of the balancing hole is increased. In the current impeller five equally spaced hole in a constant pitch circle were drilled. The drilling diameter is increased to 2mm, 3.5mm and 5mm. Figure 2 shows the impeller with drilled holes.

Figure 1. Impeller without any balancing hole.

Figure 2. Impeller with balancing hole.

4. Numerical modelling

4.1. Hydraulic domain modelling

To conduct a numerical investigation on the pump performance, the 3D models of the pump were obtained using digitized camera and modelling software of CATIA V5. Thereafter, the flow domain from these model were extracted for the next steps. **Figure 3** displays the 3D models of the casing and the impeller.

4.2. Mesh generation

The first step toward obtaining accurate results is generating a reliable grid. For generating the computational mesh, the hydraulic domains were divided into five sections (See Figure 3), namely inlet pipe, impeller, volute, side chamber and outlet pipe. The inlet and outlet pipe were designed so that fully developed flow condition is achieved in the impeller inlet and pump outlet. The mesh grid for each section was generated separately. Figure 4 show the mesh grid generated for impeller and volute. Due to complexity of the geometry, tetrahedral and hexahedral elements were used for different regions. In the vicinity of the critical regions such as leading and trailing edges or volute tongue where flow separation is anticipated, the mesh points are refined. In the procedure of mesh generation orthogonality, aspect ratio and skewness were attempted to lay within the acceptable range.

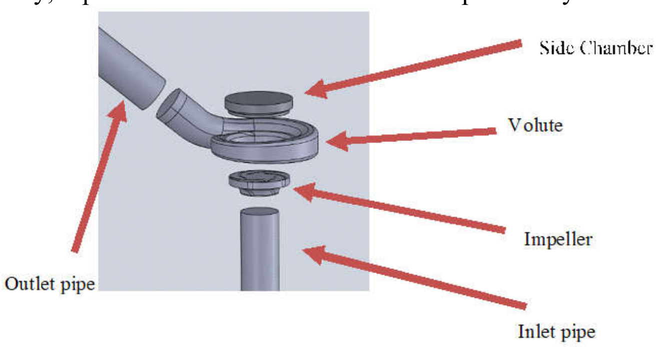


Figure 3. Hydraulic domains

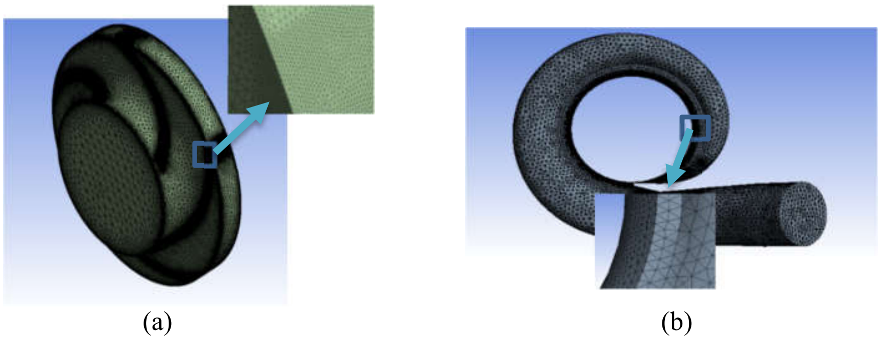


Figure 4. Mesh grid.

4.3. Numerical simulation

In the current research the commercial software of ANSYS CFX R16.0 was used to simulate the flow in the pump. The boundary condition used in the simulation are reported in the **Table 3**. As cited in the literature [14] this set of boundary conditions leads to accurate and robust results. Maximum residuals criterion for P-mass and Momentum were set to 10^{-6} and 10^{-5}. Frozen rotor type interface was chosen for inlet-impeller and impeller-volute domains. In this method, stationary coordinate are used to solve the governing equations in the stator and for the rotor, rotating coordinate is used. In the rotating coordinate, Coriolis and rotating terms are added to the acceleration terms and the rotating impeller is considered to be fixed. Therefore, in one side, the relative position of these rotating and stationary coordinates is maintained fixed and on the other side the rotation of rotor is considered. This looks like capturing a snapshot from the flow and obtaining the results for that specific time. This method can beneficially reduce the computational costs and simulation time. For the regions that two domain have the same coordinate, general connection interface is utilized.

Table 3. Boundary conditions

Parameter	Value
Inlet	Total head
Outlet	Constant outlet mass flow, normal to boundary
Inlet-Impeller interface	Frozen rotor
Impeller-Volute interface	Frozen rotor
Volute-Outlet interface	General connection
Volute-Side chamber interface	Frozen rotor
Walls	Volute: roughness 0.05, no-slip Impeller: roughness 0.02, no-slip Other: roughness 0.03, no-slip

5. Results and discussion

5.1. Grid independency

The first step in achieving a numerical result is investigating the independency of result with respect to the mesh grid. Therefore, in the case of impeller without any hole, mesh grids of fluid domains were generated. Figure 5 depicts the variation of head coefficient at BEP with respect to the grid elements. As it is shown, by increasing the grid elements from 3×10^6 to 3.4×10^6, the head coefficient varies below 0.3%. Therefore, in all test cases, the mesh with 3×10^6 elements was chosen as the optimum by which results are accurate enough without the excessive penalty of computational costs. Thereafter, for impellers with balancing holes of diameter 2, 3.5 and 5 mm the mesh grids were also generated.

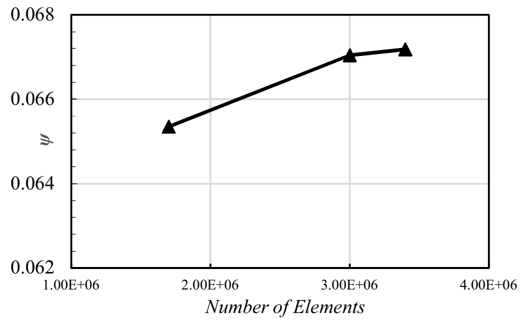
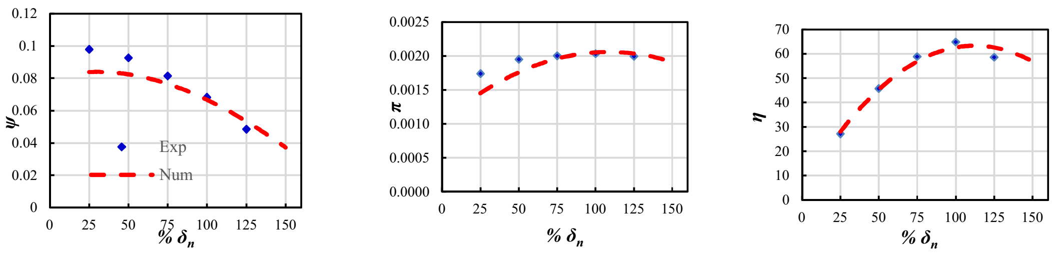


Figure 5. Grid independency.

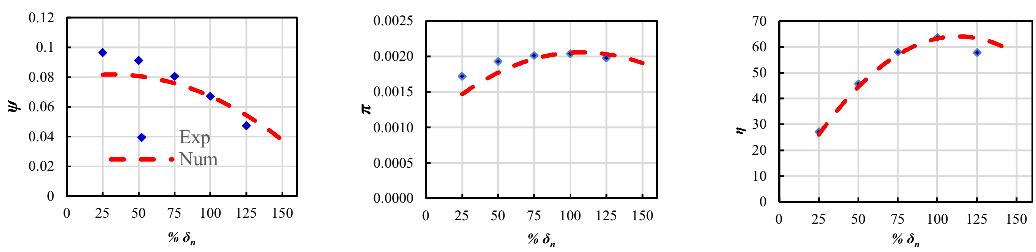


(a) (b) (c)

Figure 6. Pump characteristics curves without any balancing hole.

5.2. Performance curves.

Figure 6 displays the performance curves of pump in the case of no balancing hole. As it is shown, the head follows a monotonically decreasing trend. In other words, by increasing the flowrate, the pump head decreases. Power has an alternating behaviour meaning that from low flowrates to the nominal flow, the power is increasing. Next, by passing the best efficiency point, the power is decreased. This is the behaviour expected from pumps with medium specific speeds. Figure 6-c depicts that the efficiency, like power, does not follow a monotonic trend. It increases at first up to nominal flowrate and then it decreases. The maximum value of the efficiency in this case is 63%.

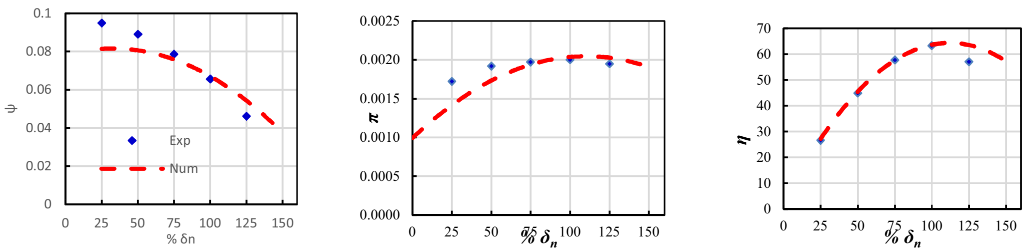


(a) (b) (c)

Figure 7. Pump characteristics curves with balancing hole of diameter 2 mm.

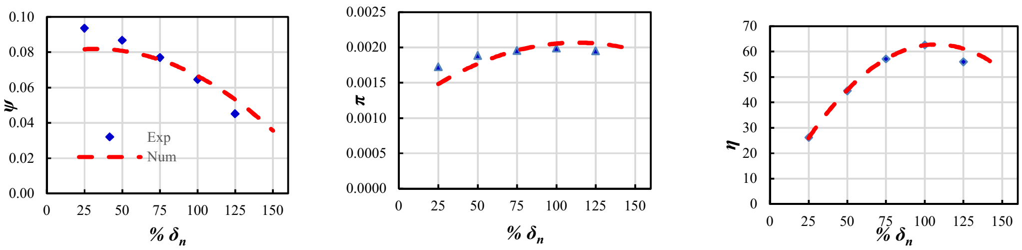
According to Figure 6, a good agreement between numerical results and experimental data is found. In the best efficiency point the relative error is below 1%. In the regions in adjacent to BEP, results are accurate, but by deviating from the BEP, the difference between numerical results and experiments increases. This can clarified in this way that in off-design conditions, the non-uniformity of flow field increases and the presence of vortices and other transient phenomenon makes it hard to reach an accurate results.

Figure 7 shows the characteristic curves for the first step of drilling, i.e. hole size of 2 mm. **Figure 7**-a depicts the head coefficient with respect to flowrate. Similar to previous step, the monotonically decreasing trend of head coefficient can be seen. Head coefficient follow a decreasing trend and by increasing flowrate up to nominal flow it takes the value of 0.067. **Figure 7**-b shows the variation of pump power with respect to flowrate. The power behaviour is identical to previous step. Again, power increases up to reaching to the nominal flowrate and then after taking its maximum, it decreases. The performance curves of the cases with impeller balancing hole diameters of 3.5 and 5 mm are shown in **Figure 8** and **Figure 9**. As it can be seen, the overall trend of all cases are the same. In the following, the effect of each balancing holes on characteristics parameters is described separately.



(a) (b) (c)

Figure 8. Pump characteristics curves with balancing hole of diameter 3.5 mm.



(a) (b) (c)

Figure 9. Pump characteristics curves with balancing hole of diameter 5 mm.

5.3. *Balancing hole diameter effects on head*

Table 4 provides the values of head coefficient for different flowrates with different balancing holes. As it shown, there are small discrepancies between numerical results and experiments. Figure 10 displays the effect of balancing hole diameter on the head coefficient according to experimental results. Vertical axis refers to the head coefficient and horizontal axis displays the hole diameter. The value of zero at horizontal axis corresponds to the case with no balancing hole. As shown, by increasing the hole diameter, the head coefficient decreases at all flowrates. This can be expected as making holes in the impeller implies extra internal leakage and larger holes results in higher leakage within the pump. Internal leakage means that a portion of power transmitted to impeller has been destroyed instead of being utilized for pressure rise. It can be understood from Figure 10 that the hole diameter is influential on the head coefficient specifically in low flowrates. Interestingly, the head at flow of 50% has been affected the most and in the worst case it has dropped about 8.6%. At the BEP, increasing the hole diameter to 2, 3.5 and 5 mm leads to 1.4%, 4.3% and 5.6% reduction of head coefficient respectively. Another point to be mentioned is that the variation of the head coefficient caused by increasing hole diameter from 2 to 3.5 mm is larger than that of increasing from 3.5 to 5 mm.

Table 4. Effect of balance hole diameter on head coefficient.

	φ=0 (mm)		φ=2 (mm)		φ=3.5 (mm)		φ=5 (mm)	
% δ_n	ψ_{Exp}	ψ_{Num}	ψ_{Exp}	ψ_{Num}	ψ_{Exp}	ψ_{Num}	ψ_{Exp}	ψ_{Num}
25	0.098	0.084	0.097	0.083	0.095	0.083	0.094	0.083
50	0.093	0.083	0.091	0.078	0.089	0.079	0.087	0.080
75	0.082	0.076	0.081	0.075	0.079	0.075	0.077	0.074
100	0.068	0.067	0.067	0.069	0.066	0.068	0.065	0.068
125	0.049	0.053	0.047	0.056	0.046	0.055	0.045	0.054
150	-	0.037	-	0.037	-	0.037	-	0.035

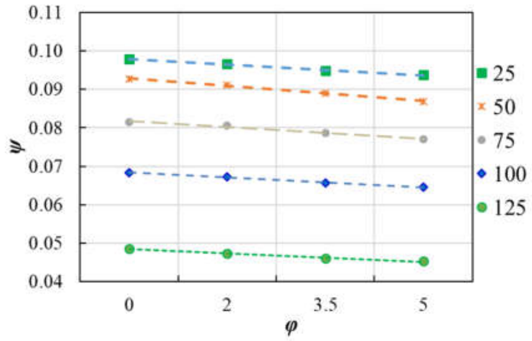


Figure 10. Balancing hole effects on head coefficient.

5.4. *Balancing hole diameter effects on power*

Experimental and numerical data of power coefficient for all flowrates with different balance holes are presented in **Table 5**. The effect of hole diameter on power based on experiments is shown in Figure 11. By increasing the flowrate the effect of balance hole diameter on power consumption seem to be more pronounced. The maximum power reduction corresponds to the flowrate of 50% of nominal flow that in the maximum hole diameter it has decreased about 3.4%. In higher flowrates, power does not exhibit a monotonic trend such a way that by increasing the hole diameter the power first increase, but after reaching the minimum point, increasing trend can be observed.

Table 5. Effect of balance hole diameter on power coefficient.

	φ=0 (mm)		φ=2 (mm)		φ=3.5 (mm)		φ=5 (mm)	
% δ_n	$\pi_{Exp}\times10^3$	$\pi_{Num}\times10^3$	$\pi_{Exp}\times10^3$	$\pi_{Num}\times10^3$	$\pi_{Exp}\times10^3$	$\pi_{Num}\times10^3$	$\pi_{Exp}\times10^3$	$\pi_{Num}\times10^3$
25	1.742	1.497	1.720	1.501	1.723	1.466	1.724	1.522
50	1.954	1.661	1.929	1.710	1.916	1.624	1.886	1.682
75	2.006	1.993	2.011	1.974	1.972	1.969	1.956	1.984
100	2.035	2.107	2.039	2.092	2.002	2.083	1.993	2.083
125	1.999	2.009	1.981	2.027	1.947	2.018	1.955	2.060
150	-	1.910	-	1.897	-	1.895	-	1.950

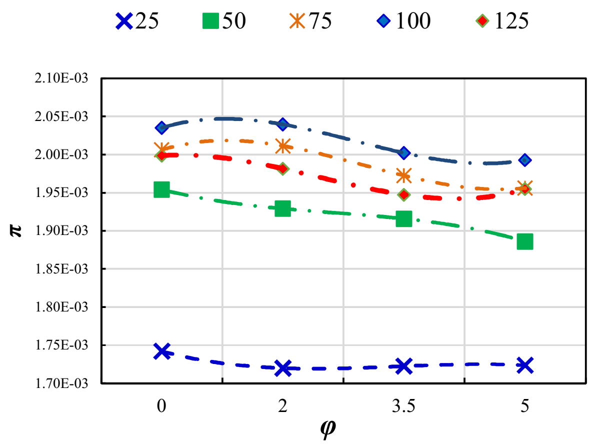


Figure 11. Balancing hole effects on power coefficient

5.5. *Balancing hole diameter effects on efficiency*

Experimental and numerical data of pump efficiency for all flowrates with different balance holes are presented in **Table 6**. Figure 12 display the effect of balancing hole diameter on pump efficiency. As clearly can be seen, the efficiency is decreased by increasing hole diameter. Balancing hole effects become more noticeable in high flowrates. At the nominal flow, balancing holes of 2, 3.5 and 5 mm yield efficiency reduction of 1%, 1.4% and 2% respectively.

Table 6. Effect of balance hole diameter on efficiency.

	φ=0 (mm)		φ=2 (mm)		φ=3.5 (mm)		φ=5 (mm)	
% δ_n	η_{Exp}	η_{Num}	η_{Exp}	η_{Num}	η_{Exp}	η_{Num}	η_{Exp}	η_{Num}
25	27.15	27.06	27.11	26.76	26.61	27.20	26.26	26.22
50	45.86	48.01	45.64	44.19	44.88	46.84	44.51	45.91
75	58.89	55.44	58.03	55.27	57.76	55.27	57.10	53.93
100	64.91	61.47	63.64	63.27	63.30	63.38	62.59	63.38
125	58.62	63.92	57.75	66.16	57.10	66.20	55.91	63.56
150	-	56.53	-	55.92	-	55.96	-	51.59

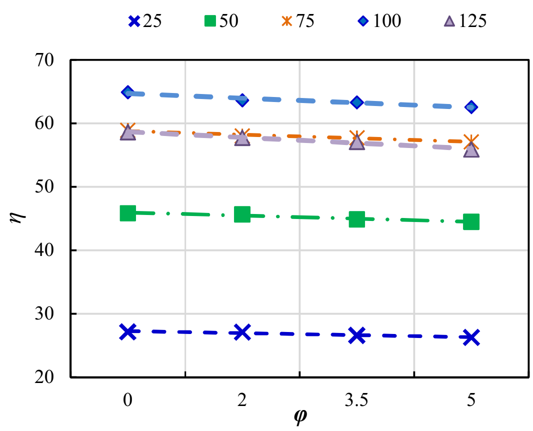


Figure 12. Balancing hole effects on efficiency.

5.6. Balancing hole diameter effects on axial thrust

After validation of numerical result with the experiments, the axial thrust for each test case for all flowrates are obtained from the result given by flow simulation. This force is the net force exerted on front and rear of the impeller. The dimensionless force is introduced as:

$$F_a^* = \frac{F_a}{\rho u_n^2 d^2} \tag{1}$$

where F_a, ρ, u_n and d are axial thrust, density, tangential velocity at BEP and impeller diameter respectively.

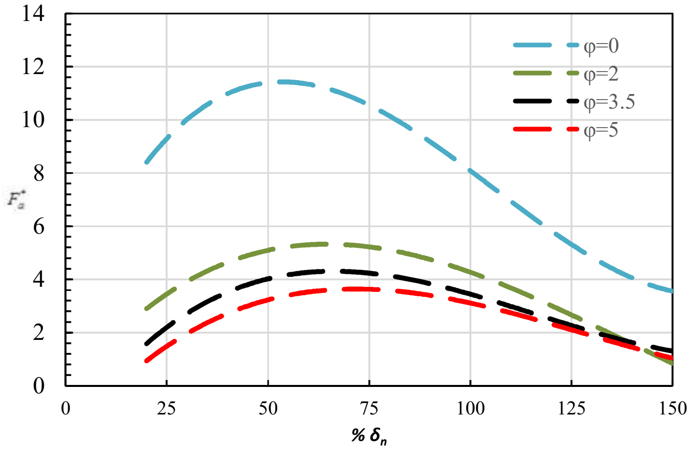


Figure 13. Balancing hole effects on axial force.

Figure 13 displays the axial thrust for each balancing hole diameter in all flowrates. As it can be seen, the axial force at first increases and after taking its peak value follows a descending trend. In the case of no balancing hole, the maximum axial thrust is obtained at 50% of nominal discharge. However, by adding balancing holes, this point shifts toward to regions close to 60% of nominal flowrate. Adding balancing holes with diameter of 2 mm has caused reduction of axial thrust in BEP about 45%.. The hole diameter of 3.5 mm has reduced the axial thrust in BEP by about 56% in comparison with no hole case. However, by increasing hole size from 3.5 to 5 mm only negligible force reduction (about 1%) is observed. It is worthy to mention that in high flowrates, adding a hole sensibly reduces the axial force in comparison to case of impeller without any hole. However, the increasing diameter of balancing does not alter axial thrust sensibly. As depicted in Figure 13, in high flowrates, axial forces approach to a same value for all balancing holes.

5.7. Balancing hole diameter effects on internal leakage

The balancing hole is a key parameter that affects the internal leakage within the pump. As a balance hole is drilled in the impeller, a new passage for transferring the fluid from the back side of the impeller to its front side is created. This new passage reduces the hydraulic resistant between two sides and consequently more flow is circulated between the sides. The occurrence of leakage flow can be interpreted as the main of cause of pump deterioration as well as force reduction. Figure 14 shows the flow streamlines in the pump with 5mm hole and without hole for 150% and 75% of nominal flow rate at a mid-plane section. As illustrated, there are streamlines that pass from impeller back side to its front side through the balance hole. The amount of flow passing through the hole is related to the pressure distribution between two sides of the impeller which is consequently related the flowrate and the hole diameter.

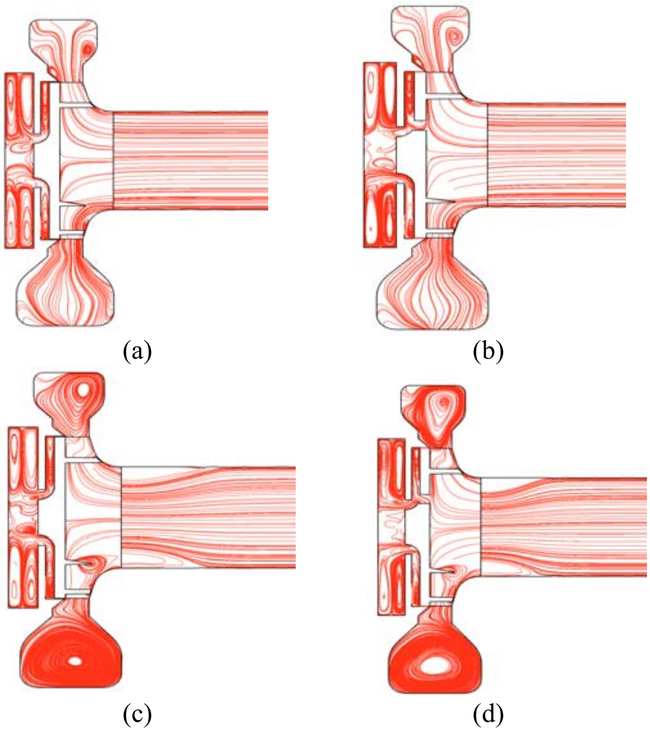


Figure 14. Flow streamlines around the impeller: a: 75% of nominal flow and no hole, b: 75% of nominal flow, hole of 5mm, c: 150% of nominal flow and no hole, d: 150% of nominal flow, hole of 5mm.

As it is shown in Figure 14, by drilling the impeller, the fluid flow in the front of impeller is less affected than the flow in the back side of the impeller. Another important finding is that when a hole is provisioned in the impeller, the strength of the vortices in the impeller back side is reduced and some of them are disappeared.
For analyzing the leakage flow quantitatively, a dimensionless parameter of ε is defined as:

$$\varepsilon = \delta_l/\delta_n \times 100 \tag{2}$$

Which shows the percentage ratio of leak flow to design flowrate.

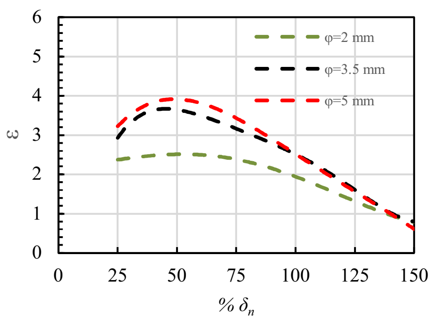


Figure 15. Effects of balancing holes on the pump performance

Figure 15 displays the effect of balancing hole on the leak flow within the pump. As it is shown, the maximum leakage corresponds to the 50% of nominal flowrate and by increasing the flowrate above it, the leakage is reduced. In the case of no hole, the leakage flow is about zero and as it is expected, by increasing the hole diameter, the leakage flow increases. Numerical results show that using hole diameter of 2 mm increases leakage flow considerably. Increasing hole diameter more mostly increase leakage flow in the low flowrates. However, hole diameter size becomes less effective in high flowrate

As it is depicted, drilling hole of diameter 2mm can increase leakage flow up to 2% in the nominal flow, and by using larger holes, leakage flow approaches to 2.6%.

6. Conclusion

In this paper, the effects of balancing hole diameter on the pump characteristic as well as axial thrust were studied numerically and experimentally. It was found that the effect of hole diameter on head is remarkable at low flowrates. The flowrate of 50% which is affected the most, experienced head reduction of 8.6%. The efficiency was also found to be affected negligibly with the hole diameter. It was observed that the axial force is highly correlated with the hole size, and by making balancing holes of 3.5 mm the axial thrust is reduced about 56%. By increasing the hole diameter, the axial force is reduced but the variations are not noticeable. Therefore, in this research the hole size of 3.5 mm is proposed for the current test case because the axial force is reduced acceptably and deterioration of pump performance are within reasonable range.

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