IMPROVEMENT OF CENTRIFUGAL PUMP PERFORMANCE BY USING DIFFERENT IMPELLER DIFFUSER ANGLES WITH AND WITHOUT VANES

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ABSTRACT

This study aims at improving the performance of a centrifugal pump by using different angular diffusers on the downstream side of the centrifugal pump impeller. Numerical and experimental studies have been carried out on different vaned and non- vaned diffuser with three different wall divergence angle (α) of 0°, 5° and 10° to achieve that purpose. The data analyses show good agreement between the numerical and experimental results. They reveal profound effect of the divergence angle (α) of angular vaned diffuser on the head and overall efficiency of centrifugal pumps especially at high flow rates as they broaden operating region of the centrifugal pump. In fact it is found that the head and overall efficiency of impeller with vaned diffuser $\alpha = 10°$ enhance by 15.4 and 9 percent respectively compared to that of centrifugal pump with no vaned diffuser at high flow rates. Furthermore the head and overall efficiency of impeller with vaned diffuser $\alpha = 10°$ increase by 5.7 and 7 percent respectively in comparison with the impeller with vaned diffuser $\alpha = 0°$.

Keywords: Centrifugal pump, Performance enhancement, Angular vaned diffuser, Angular non- vaned diffuser.

1. INTRODUCTION

Due to the restriction of energy resources and rising energy consumption in various kinds of industries on the whole world, optimizing of different equipments is a necessity. Among a wide range of equipments centrifugal pumps is an inevitable part of almost all industries ranging from the energy generation and the refinery industries to the chemical and the food industries. Therefore numerous scientists have analysed centrifugal pumps experimentally, theoretically and numerically to approach that aim [1-5]. As a matter of fact there are different effective methods to enhance the performance of centrifugal pumps which have been used in various researches that some of them are mentioned here.

Wang *et al*. [6] conducted a study on optimal design based on the combined numerical analyses and energy loss model (ELM/CFD). Different turbulence models, grid numbers, surface roughness and convergence precisions were considered. The ELM/CFD method was presented comprising different energy losses in the pump, in terms of volumetric leakage loss, disk friction loss, inter-stage leakage loss and the hydraulic loss.

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Their results show that to enhance efficiency of centrifugal pumps declining the volumetric leakage and inter-stage leakage losses is the most effective method. Furthermore designing impeller with regard to maximum flow rate due to the inevitable volumetric leakage loss will result in an efficiency enhancement at lower flow rates. Zhang *et al*. [7] by data-mining methods tried to modify a water pumping system. Energy saving of 6 to 14% by making balance between the energy cost and wastewater flow rate was achieved from their investigations. Zhang *et al*. in another work [8] performed a research on reducing pump energy in a wastewater processing plant by neural network algorithm. Their scheduling model was a mixed-integer nonlinear programming problem (MINLP). The modified results revealed that based on the computed schemes the performance of the pump system can be considerably enhanced. Impeller trimming method has been used experimentally by Savar *et al*. [9] and Singh *et al*. [10] to improve energy efficiency of centrifugal pumps. They reported that by trimming impeller to the appropriate size energy saving is accomplished. Khoeini *et al*. [11] analyzed the effects of changing of the volute throat area on the flow characteristics of an over hung centrifugal pump with viscous fluids. For diffusion and convective terms central difference and upwind (2 order) methods were used in computations, respectively. Also, they considered SST turbulence model (Shear Stress Transport) for simulations. Their results demonstrated that the volute throat area has profound influence on the head of centrifugal pumps as they achieved a head rise of 6.4% in their studies. Dong *et al*. [12] surveyed optimization influences of tongue and impeller geometry on a centrifugal pump experimentally. They utilized Particle Image Velocimetry (PIV), pressure, and noise measurements to find the influence of improvements to impeller geometries and tongue on the centrifugal pump performance. They accomplished to the lowest noise by increasing the gap, approximately 20% of the impeller radius, between the impeller and the tongue. The stiffness of the tongue is also found that as an important issue for noise reduction. Zhu *et al*. [13] studied the influences of half vane diffuser on the performance of a centrifugal pump numerically and experimentally. They considered 3-D, unsteady incompressible Reynolds- averaged Navier- Stokes equations to analyze the inner flow field by the commercial CFD software ANSYS- FLUENT in a segregated case with SIMPLE algorithm. Finite-volume method was employed for discretization of governing equations and second-order upwind discretization scheme was utilized for time-dependent terms. An efficiency increase of 2.5% in design flow point by using the half vaned diffuser to the centrifugal pump was obtained in their studies. Stel *et al.* [14] numerically investigated the flow field in the first stage of a centrifugal pump with a vaned diffuser. Computations were carried out by using the CFX software from ANSYS. The flow field was modeled by the unsteady Reynolds-averaged Navier-Stokes (U-RANS) equations as the standard k-ε turbulence model was regarded. Substantial levels of turbulence and blade orientation influences in the diffuser were observed in their results. Besides, by keeping the flow dimensionless coefficient constant the flow pattern was similar in different capacities and rotor speeds. Gaetani *et al*. [15] analysed the effects of different vane setting angles of a vaned diffuser on the centrifugal pump low field with 2-D-PIV measurements in various blade to blade planes. The diffuser setting angle affected the spanwise distribution of the absolute mass average flow angle at discharge of impeller and the flow rate. Also in the two highest setting angles their results demonstrated an overturning at the diffuser discharge. Zhou *et al*. [16] carried out a numerical and experimental study on a deep-well centrifugal pump with different diffusers. Five different turbulence models were evaluated including the standard k-ε model, RNG k-ε model, realizable k-ε model, standard k-w model, and SST k-w model. Based on data analyses the standard k-w model was employed in 3D incompressible steady flow of the commercial code ANSYS-Fluent 13.0. The diffusive and convective terms schemes were second order upwind. Moreover, second order implicit was regarded for the time dependent term scheme. In all operating conditions they accomplished better results by three-dimensional surface return diffuser rather than that for the traditional cylindrical return diffuser. Atif *et al*. [17] surveyed the vaned diffuser interaction inside the velocity field of a centrifugal pump impeller numerically and experimentally. Three-dimensional, steady incompressible Reynolds averaged Navier-Stokes were computed utilizing CFX 12.1 CFD code. The k-ω based SST turbulence model was chosen in analyses as well. The unsteady behavior of the flow because of the rotor-stator interaction was observed in their surveys as it affects velocity, pressure, torque and flow rate fluctuations. Goel, *et al*. [18] conducted a research to improve the hydrodynamic performance of vanes of diffuser by shape optimization. The NASA PHANTOM code was used to investigate flow field. Three-dimensional, incompressible, unsteady, Navier – Stokes code were solved. Also, the viscous and inviscid fluxes were discretized by a second-order central differencing and third-order spatially accurate Roe's scheme, respectively. They found that modification of the shape of vanes can reduce the total pressure losses by 8% in comparison with the baseline design by declining the base separation. Furukawa *et al*. [19] investigated pressure fluctuation of a centrifugal pump impeller that equipped with vaned diffuser experimentally. Rising pressure fluctuations in the vaned diffuser was observed due to the circumferential blade surface of the impeller with an outlet edge of the pressure surface. Agrawal *et al*. [20] implemented an empirical study on the rotor- stator interaction in a centrifugal pump experimentally. It is obtained that growing the vane number of diffuser leads to a substantial decline of the impeller blade fluctuations. According to conducted researches on impeller diffusers, the operating region of centrifugal pumps is broadened

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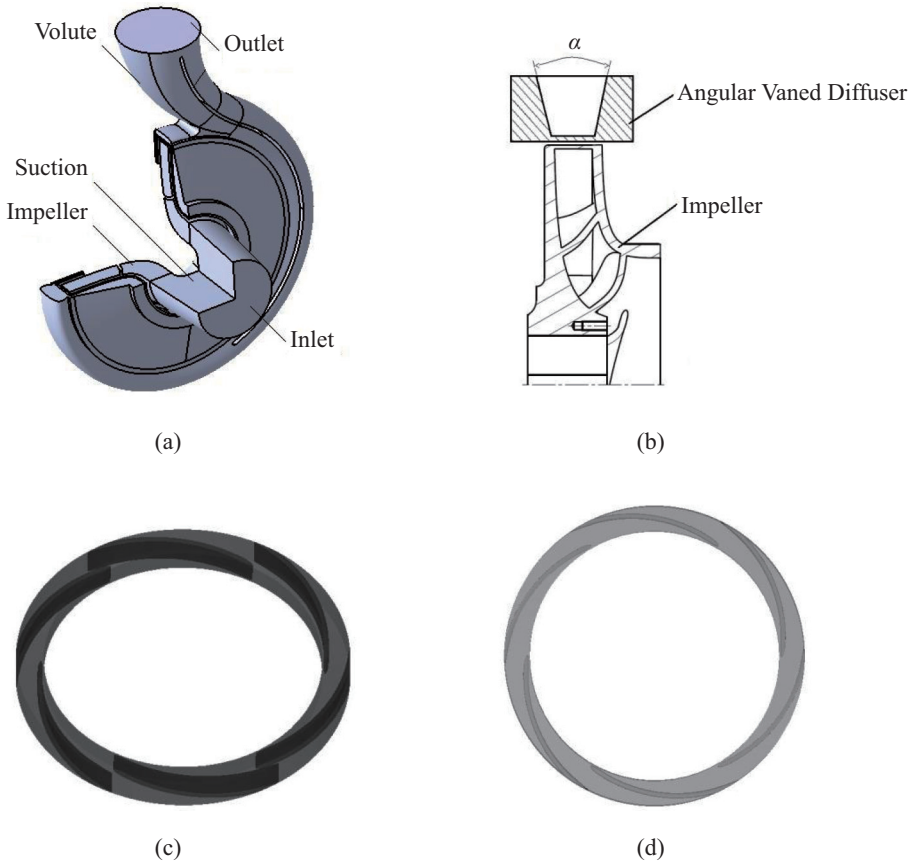


Fig. 1 (a) Centrifugal pump parts (b) Sectional view of the impeller and angular vaned diffuser (α: wall divergence angle of diffuser) (c) Isometric view of the impeller diffuser (d) Front view of the impeller diffuser.

thanks to the application of impeller diffusers. Besides, they are economical since there is no need for changing of mould of casing volute of centrifugal pumps, extra processes and other aspects accordingly.

Although many researches have been conducted on centrifugal pump optimization, but there are some studies which have been concentrated to survey the effects of impeller diffusers on the performance of centrifugal pumps but the analysis of influences of the divergence angle (α) of main body walls of impeller diffuser on the flow characteristics of centrifugal pumps has not been considered to the knowledge of the authors. In the other words, manipulating of different parameters of impeller diffusers have been done experimentally, theoretically and numerically in previous investigations but the impact of the divergence angle of main body walls of impeller diffusers (α) has not been taken into account, yet. Therefore, the present research is aimed for studying the effects of wall divergence angle (α) of 0°, 5° and 10° of vaned and non- vaned diffuser on the performance of a centrifugal pump numerically and experimentally.

2. CENTRIFUGAL PUMP PARTS

In Fig. 1(a) different parts of the centrifugal pump including inlet region, impeller and double volute casing type for minimizing radial thrust [21-22] are illustrated. Also Fig. 1(b) depicts sectional view of impeller and angular vaned diffuser which was set firmly to the volute casing by bolts. The shape of the diffuser blades remained the same for investigation of the impacts of different diffuser wall angles on the pump performance at equal conditions Figs. 1(c) and (d). In Fig. 1 wall divergence angle (α) of angular diffuser that was considered from 0° to 10° in this study is demonstrated. Each diffuser, vaned diffuser α = 0°, non- vaned diffuser α = 0°, vaned diffuser α = 5°, non- vaned diffuser α = 5°, vaned diffuser α = 10°, non- vaned diffuser α = 10°, has been studied in computational flow domain separately and based on the results from computational fluid analyses optimum impeller diffuser shape was constructed and tested thoroughly. In addition all specifications of the centrifugal pump including diameters of suction and discharge pipes, diameters of impeller and impeller diffusers, blade and vane numbers, leading and trailing angle of blades, height and thickness of blades and revolution speed are mentioned in Table 1. In fact, the surveyed centrifugal pump exists with operating condition of H = 98 m at Q = 560 m^3/h for impeller diameter of 580 mm (maximum diameter) and since impellers trimming is a common practice for meeting related applications, in this study we regarded a trimmed impeller (with diameter of 500 mm) and by equipping angular impeller diffuser higher performance has been accomplished.

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Table 1 Centrifugal pump specifications (Specific Speed = 18.4).

Impeller	Impeller Dia.	500 mm
	Inlet blade angles	19°
	Outlet blade angles	27°
	Number of blade	4
	Blade height	38 mm
	Blade thickness	8 mm
Diffuser	Diffuser inner Dia.	504 mm
	Diffuser outer Dia.	604 mm
	Number of vane	6
	Base circular width	42 mm
Volute casing	Suction pipe dia.	200 mm
	Discharge pipe dia.	200 mm

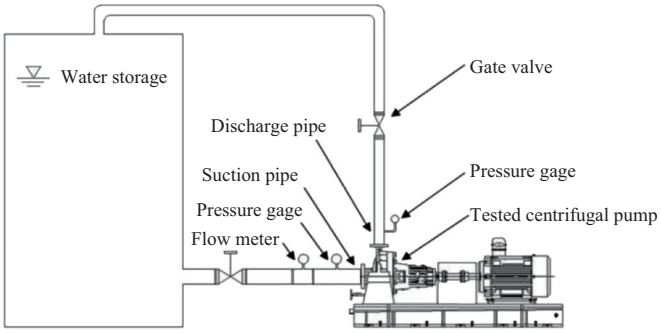
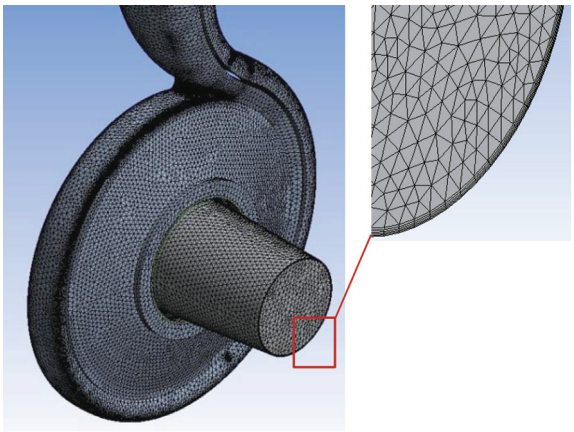


Fig. 2 Test facilities.

3. TEST FACILITIES

Figure 2 shows different parts of the test rig that have been used in the present experimental investigation. Water at 25°C has been considered as working fluid [23]. The centrifugal pump was driven by an electrical AC driver, 1475 rpm. By making changes of the discharge valve from shut off to fully open all performance curves of the centrifugal pump, differential head against flow rate, overall efficiency against flow rate, have been obtained. Furthermore, power has been measured with a torque meter in each flow rate as well. Various flow rates and pressure values of suction and discharge pressure gages were measured according to [24]. Uncertainty values of head, flow rate and absorbed power have been calculated ±2.8%, ±2.2%, ±2.9% respectively based on [25]. The main flow parameters of the original pump including head coefficient, flow coefficient and Reynolds Number are 0.12, 0.005 and 57.5 × 10^6 respectively.



(a)

Grid size (×1000)	344	572	1009	1797
Torque (N.m)	793	844	875	882

(b)

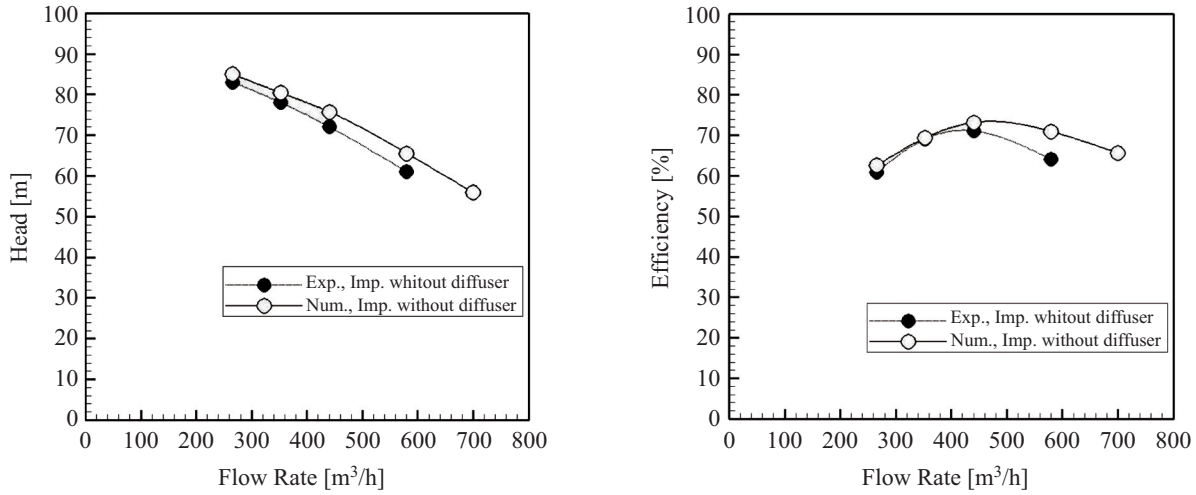
Fig. 3 (a) Computational grid for flow domain (b) The elements analysis for impeller.

4. COMPUTATIONAL ANALYSES

4.1 Mesh Generation and Verification

Figure 3 illustrates computational grid for flow domain. Elements have been refined in all the near acute regions of the computational domain. Structured mesh has been employed in the near wall regions and unstructured mesh has been used in the regions far from the walls. In the areas between rotating part (impeller) and nonrotating parts (suction region and volute casing region) separable interfaces have been used. To minimize the process

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(a) (b)

Fig. 4 (a) Head as a function of flow rate, experimental and numerical results (b) Efficiency as a function of flow rate, experimental and numerical results.

time and increasing the precise of computation the optimum number of elements have been investigated. As shown in Fig. 3(b), surveys have been carried out for becoming independent from dimensions of elements. Eventually after different iterations, the minimum error of less than 1 percent was accomplished and employed in this study (the grid size of 1797 × 1000 cells). Additionally, y+ value for all cases is lower than 50 which leads to higher resolutions on the boundary layers.

4.2 Governing Equations

In the present steady state study because of the incompressible working fluid, the Navier-Stokes equations, continuity and momentum equations have the following forms [26]:

$$\nabla \boldsymbol{U} = 0 \tag{1}$$

$$\rho \nabla.(\boldsymbol{UU}) = -\nabla p + \nabla.(\tau) + \rho \boldsymbol{g} + \boldsymbol{F} \tag{2}$$

where ρ, U, p, τ and F present fluid density, velocity, pressure, stress tensor and source term (centrifugal force) respectively. Central difference discretization for diffusion terms and upwind (2 order) for convective terms have been used in computational analyses. For inlet boundary condition, the total pressure with uniform flow and turbulent_intensity of 5% has been specified. For outlet boundary condition, mass flow rate has been specified. At all physical surfaces of regions of suction, impeller, diffuser and volute the no slip boundary condition has been considered. And SST turbulence model (Shear Stress Transport) has been employed as validating of its accuracy has been done in numerous investigations [27]. Actually, in the stated turbulence model, a combined functions of standard k-ω and k-ε turbulence models are applied in the immediate vicinity of wall and in the fully developed region, respectively. Since transmitting of turbulence shear stress is regarded, more precise results close to wall are accomplished and flow separation is predicted more accurately.

Moreover, frozen rotor approach has been employed for flow simulations which regards the rotor as stationary and the rotational influences are considered as source term in the momentum equations.

4.3 Comparison between Computational and Experimental Results

At first centrifugal pump without impeller diffuser has been studied numerically and then it has been tested according to [23] thoroughly (Fig. 4). As it is observed the agreement between the computational and experimental results is reasonably good as the highest head and efficiency discrepancy are 7.3 and 6% respectively at the same operating conditions (inlet uniform flow, inlet pressure, working fluid).

5. RESULTS

As mentioned above, in this research three- dimensional computation domain of a centrifugal pump has been studied for regarding different aspects of flow such as secondary flows in analyses. Investigations on the three different vaned and non- vaned diffusers $\alpha = 0°$, $\alpha = 5°$, $\alpha = 10°$ have been performed thoroughly and the optimum impeller diffuser shape was constructed and mounted for experimental tests.

Variation of head against flow rate for pump without diffuser and with vaned and non- vaned diffuser of $\alpha = 0°$ is given in Fig. 5(a). As it can be seen, the head and efficiency of the impeller with vaned diffuser $\alpha = 0°$ are higher than those of the impeller without any diffuser. At the high flow rate, the maximum head rise is by 8.9%. While for the impeller with non-vaned diffuser $\alpha = 0°$ and the impeller without diffuser have the same head

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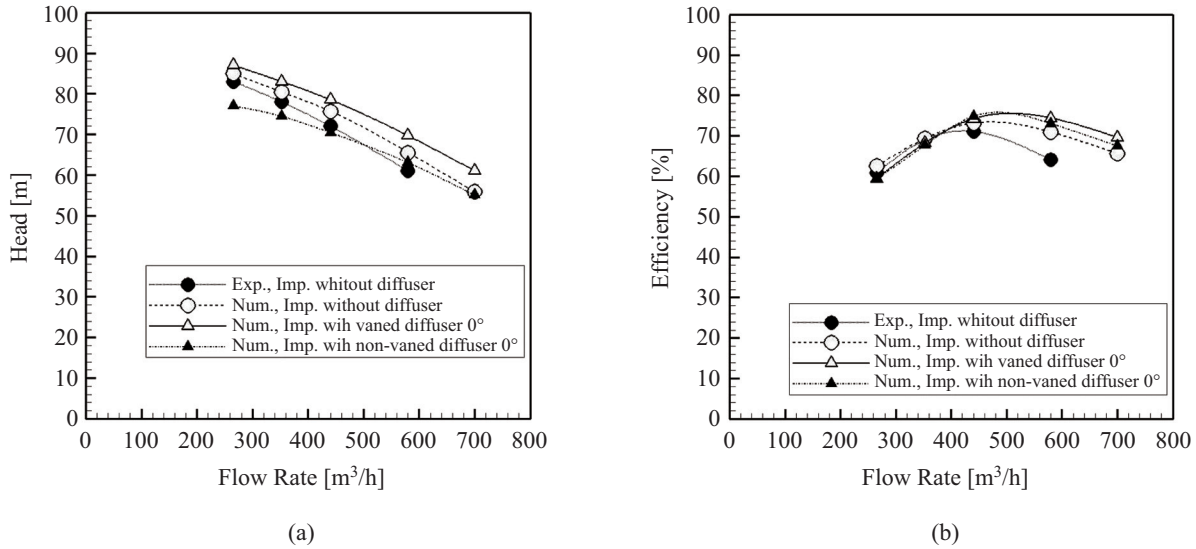


Fig. 5 Impeller with diffuser $\alpha = 0°$ (a) Variation of head against flow rate (b) Variation of overall efficiency against flow rate,

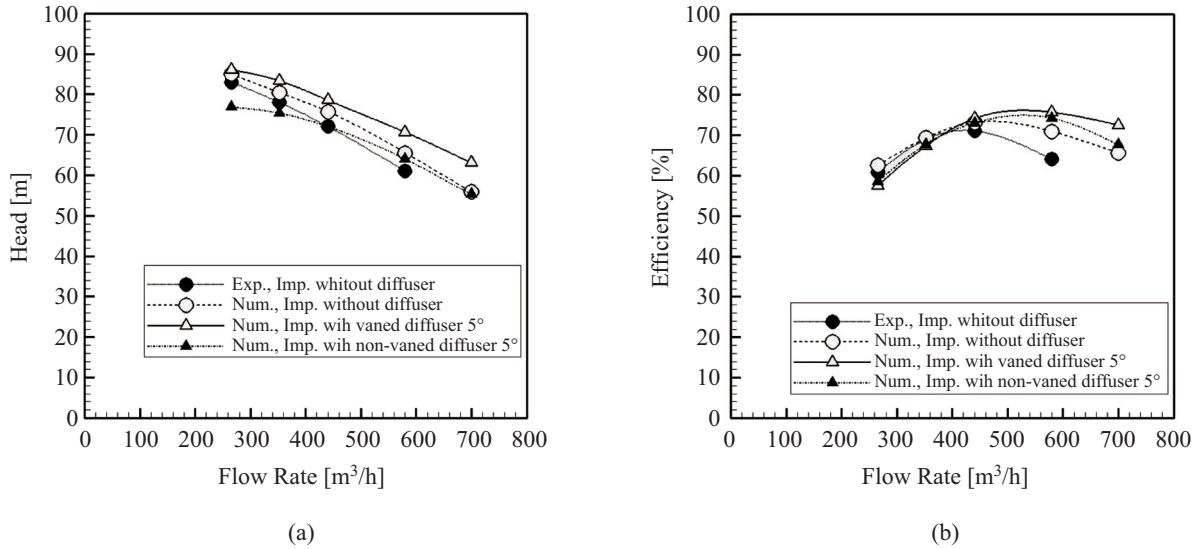


Fig. 6 Impeller with diffuser of $\alpha = 5°$ (a) Variation of head against flow rate (b) Variation of overall efficiency against flow rate.

values at the high flow rate but their head differences increases by 8.5% at low flow rate.

Figure 5(b) depicts the variation of overall efficiency versus flow rate for pump without diffuser and with vaned and non- vaned diffuser of $\alpha = 0°$. Comparison between experimental and numerical efficiency results of the centrifugal pump without diffuser presents good agreement and maximum deviation between them is 6% at highest flow rate. By using diffuser $\alpha = 0°$ the Best Efficient Point, BEP, is displaced towards the high flow rate, about 10.8%, due to the diffuser effects and the overall efficiency of impeller with vaned or non- vaned diffuser $\alpha = 0°$ enhanced by 2.4 and 1.1% respectively in comparison with the impeller without any diffuser at BEP. Additionally at low flow rate they have the same efficiency as well.

The effects of vaned and non- vaned diffuser with $\alpha = 5°$ on the head and overall efficiency of the centrifugal pump are depicted in Figs. 6(a) and (b) respectively. As shown by diverging the main body walls of vaned diffuser to $\alpha = 5°$ the head rises by 12.5% compared with the impeller without diffuser at high flow rates but it become close to the head without diffuser at low flow rates. On the other hand, the impeller with non- vaned diffuser $\alpha = 5°$ has the same head with the impeller without diffuser at high flow rates but some differences are observed at low flow rates. Based on the results, the divergence angle (α) of the diffuser has a beneficial effect on the overall efficiency as the flow rate increases the higher efficiency values achieve and the overall

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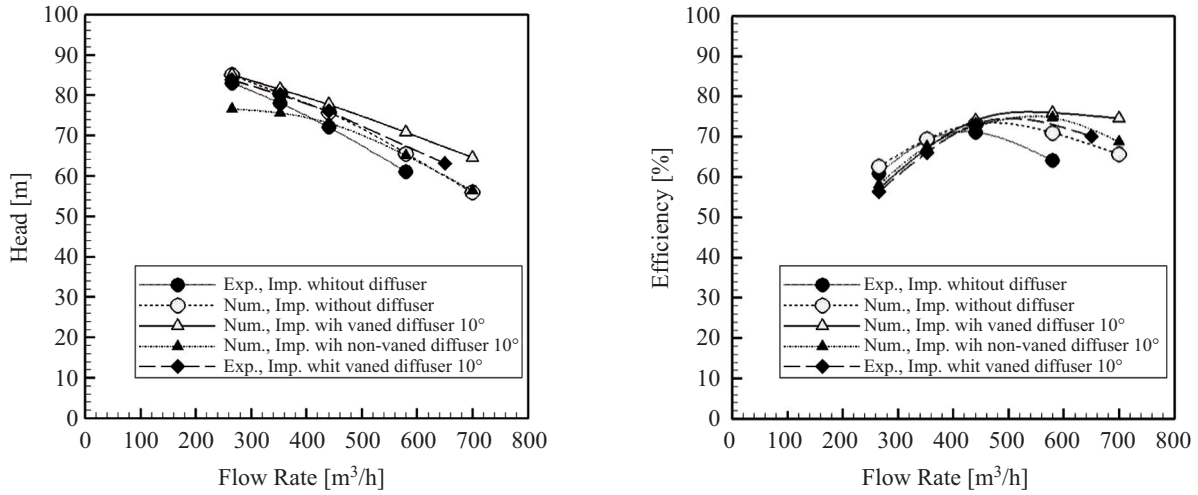


Fig. 7 Impeller with diffuser of α = 10° (a) Variation of head against flow rate (b) Variation of overall efficiency against flow rate.

efficiency enhanced by 6.8% and 2.1% for impeller with vanded and non- vaned diffuser α = 5° respectively in comparison with the impeller without diffuser. It is because of the better changing of kinetic energy to potential energy owing to the more passage area of angular diffusers rather than that for diffuser α = 0° at high flow rates. In addition like the trend of the overall efficiency curve of diffuser α = 0°, the BEP of diffuser with α = 5° moved by 14.1% towards high flow rates.

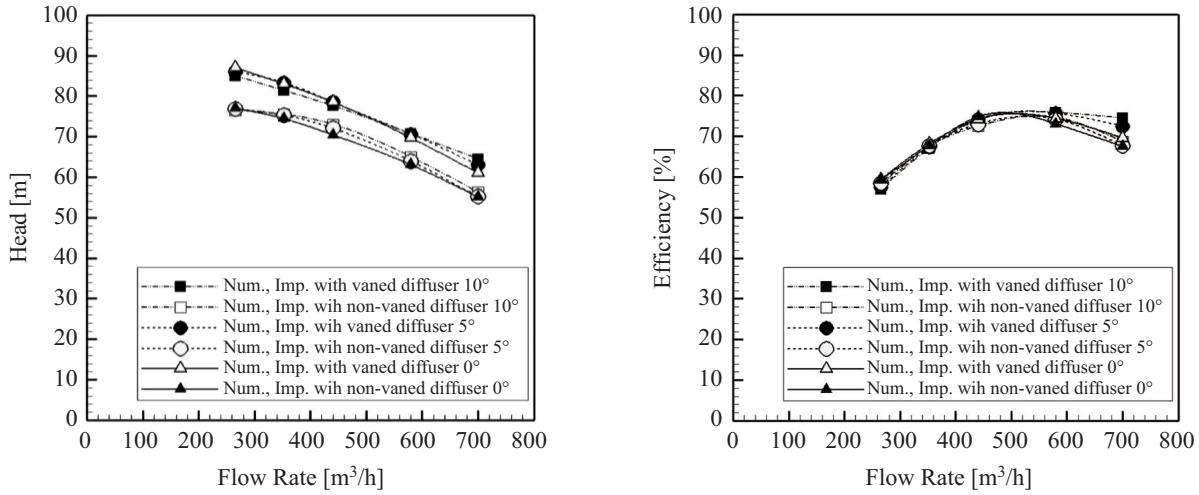
Figure 7(a) demonstrates the head variation of the pump equipped with vaned and non- vaned diffuser of α = 10° against the flow rate. Comparing of the experimental and numerical results of vaned diffuser with α = 10° showed good compatibility and maximum deviation among them is 6.6% at the high flow rate. It is observed that at high flow rate impeller with vaned diffuser of α = 10° provides higher head, by 15.4%, with respect to the impeller without diffuser and at low flow rates the same head values are achieved. The reason is that by diverging the vaned diffuser, more space are created for outgoing flow therefore there is less resistance on the flow rate and higher head values are obtained. Actually, angular vaned diffuser helps diffuse the velocity of outgoing flow from the circumferential edge of impeller and consequently conversion of the kinetic energy to the static pressure is increased significantly. Furthermore, by using vaned diffuser with α = 10° broadened operating region has been accomplished as well. The impeller with non- vaned diffuser of α = 10° has negligible head differences with that of impeller without diffuser at high capacity but at low capacity ($Q < Q_{BEP}$) it has lower head values.

Overall efficiency of impeller with vaned and non-vaned diffuser α = 10° is depicted in Fig. 7(b). Based on the data analyses the angular diffusers have positive effect on the centrifugal pump efficiency since they provide more passage area for the flow at their discharge area on the upstream side of the volute casing with respect to the diffusers with α = 0°. It is found that efficiency of the impeller with vaned and non- vaned diffuser of α = 10° are 9 and 3.2 percent higher than that of impeller without diffuser respectively at high flow rate. But their efficiency are 5.5 and 4 percent lower than efficiency value of the impeller without diffuser at low flow rates. Besides BEP position of the impellers with vaned and non- vaned diffuser shifted toward the high flow rates by 17.4% because the divergence angle of diffuser of α = 10° leads to differing energy conversion (kinetic to potential). Furthermore, increase of deviation between numerical and experimental results with flow rate as seen in Figs. 4 to 7 is due to some inseparable issues on experimental tests such as liquid passes within the internal running clearances of the centrifugal pump.

Variation of head and efficiency against flow rate at different diffuser wall angles (α) have been demonstrated in Fig. 8(a) and Fig. 8(b), respectively. As it is shown in Fig. 8(a), by enhancing α higher head values are resulted at high flow rates while by moving toward the minimum flow rate the head either remains constant or reduces slightly, and consequently leads to smoother curves. The first one was observed for non- vaned diffusers and the latter was seen for vaned diffusers. Additionally, by increasing α higher efficiency values are obtained at high flow rates as diverged diffuser walls contribute to lower interaction between the impeller blades and diffuser. Is due to the fact that in this case larger flow passage is obtained. As data analyses illustrated, the increase of the angle of diffuser walls positively affects the pump parameters. On the whole, resulted head and efficiency values of all cases are presented in Tables 2 and 3, respectively. However, increasing the angle of the diffuser walls should be further studied as higher angles than a certain value may cause efficiency drop due to the flow separation and flow blockage.

Pressure distributions of the investigated impellers at

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(a) (b)

Fig. 8 (a) Variation of head against flow rate for different α (b) Variation of overall efficiency against flow rate for different α.

Table 2 Resulted head at different cases (m).

Case	Q = 265	Q = 352	Q = 440	Q = 580	Q = 700
Num., Imp. with vaned diffuser 10°	85 m	81.4 m	77.6 m	70.7 m	64.5 m
Num., Imp. with non-vaned diffuser 10°	76.5 m	75.5 m	73 m	65 m	56.3 m
Num., Imp. with vaned diffuser 5°	86 m	83.3 m	78.5 m	70.6 m	63 m
Num., Imp. with non-vaned diffuser 5°	76.8 m	75.3 m	72 m	64 m	55 m
Num., Imp. with vaned diffuser 0°	87 m	83 m	78.5 m	69.7 m	61 m
Num., Imp. with non-vaned diffuser 0°	77 m	74.4 m	70.4 m	63 m	55 m
Num., Imp. with no diffuser	85 m	80.4 m	75.7 m	65.5 m	55.9 m
Exp., Imp. with no diffuser	83 m	78 m	72 m	61 m	-

Table 3 Resulted efficiency at different cases (%).

Case	Q = 265	Q = 352	Q = 440	Q = 580	Q = 700
Num., Imp. with vaned diffuser 10°	57	67	74	75.8	74.5
Num., Imp. with non-vaned diffuser 10°	58	67.5	72.5	74.6	69
Num., Imp. with vaned diffuser 5°	57.5	67.3	74.1	75.6	72.5
Num., Imp. with non-vaned diffuser 5°	58.5	67.7	73	74	67.6
Num., Imp. with vaned diffuser 0°	59.3	67.6	74.8	73	67.5
Num., Imp. with non-vaned diffuser 0°	59.4	68	74.2	74.3	69.5
Num., Imp. with no diffuser	62.5	69.3	73	71	65.5
Exp., Imp. with no diffuser	60.8	69	71	64	-

the mid-height of the impeller are presented in Fig. 9. As shown, the original centrifugal pump without diffuser has higher pressure gradient and by using diffuser it reduces especially by using angular diffusers (α = 5°, α = 10°). Indeed increasing the angle of diffusers affects the distribution of pressure in outlet region of impeller as the more uniform pressure distribution has been achieved by using vaned diffuser of α = 10°. As shown in Figs. 9(b) and (c), the pressure distribution with vaned and non-vaned diffuser α = 0°, the minimum passage area in comparison with angular diffusers causes enhancing of the velocity values considerably that result in nonuniform pressure distribution in peripheral region of the impeller. But by diverging the diffuser angle, α, such effect disappears. As enhanced flow passage of angular diffusers makes interaction among impeller blades and diffuser weaker consequently and more similar uniform pressure distribution is obtained at impellers with angular vaned and non- vaned diffuser.

Figures 10 shows stream function of impeller with various diffusers in the present study. Stream function of the impeller without diffuser is shown in Fig. 10(a).

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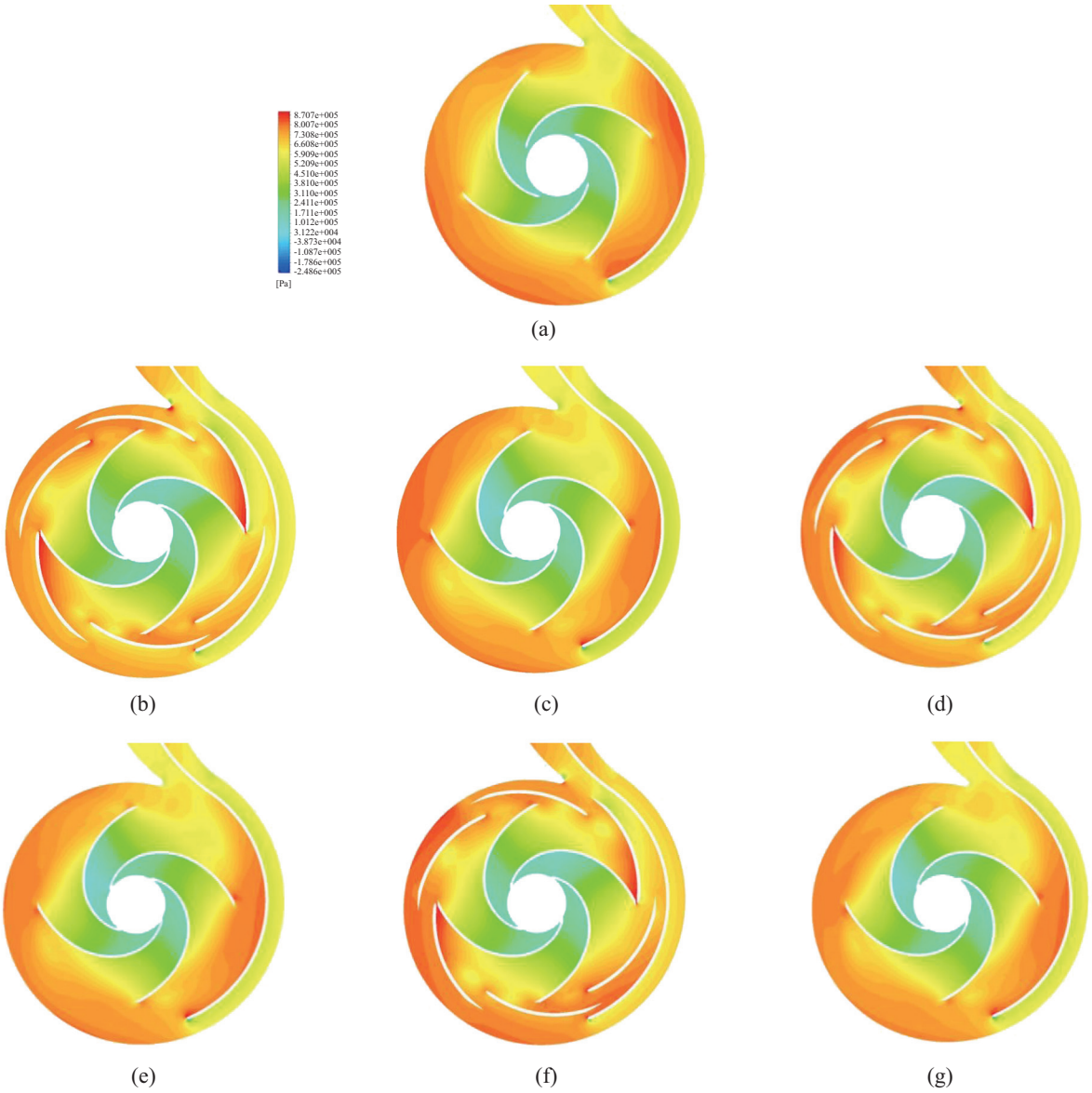


Fig. 9 Pressure distribution (a) impeller without diffuser (b) impeller with vaned diffuser $\alpha = 0°$ (c) impeller with non-vaned diffuser $\alpha = 0°$ (d) impeller with vaned diffuser $\alpha = 5°$ (e) impeller with non- vaned diffuser $\alpha = 5°$ (f) impeller with vaned diffuser $\alpha = 10°$ (g) impeller with non- vaned diffuser $\alpha = 10°$.

As it can be seen there are higher velocity gradient in the peripheral region of the impeller with respect to the impellers with vaned or non- vaned diffuser. Stream function of impellers with vaned and non- vaned diffusers of $\alpha = 0°$ are depicted as well. It is observed that due to the diffuser of $\alpha = 0°$ there is no sign of higher velocity gradient on the circumferential edge of impeller and also there are more uniform flow field on the impeller with vaned diffuser of $\alpha = 0°$ in comparison with the impeller with non- vaned diffuser of $\alpha = 0°$. Furthermore the constant passage area through these types of diffusers caused higher velocity gradients rather than that of angular diffusers with $\alpha = 5°$ and $\alpha = 10°$. Another point is that a re-circulation is seen among impeller blades with non- vaned diffusers, on the pressure side, as the difference in the flow and blade angle is the cause of the formation of re- circulation gone. While by adding vanes to the diffuser, the mentioned re- circulation zone is weakened and by enlarging the divergence angle of diffuser (α) more smooth velocity distribution on the impeller with vaned diffuser of $\alpha = 10°$ is obtained. In the other words, in contrast to the velocity distribution of impeller with diffuser with $\alpha = 0°$, at higher α values, the bulk flow becomes uniform through the passage area among impeller blades owing to the higher flow passage area of the angular diffuser. Additionally, the blockage impact of volute tongue and diffuser vanes at high flow rates affects the outgoing flow rate consequently the higher velocity gradient is observed (Figs. 10(b), (d), (f)). Also due to the narrow distance between the volute tongue and vanes of vaned diffusers, some additional wakes are caused on the vicinity of volute tongue but as wakes do not have enough strength, they have no significant impact on the local pressure, (Figs. 9(b), (d), (f)).

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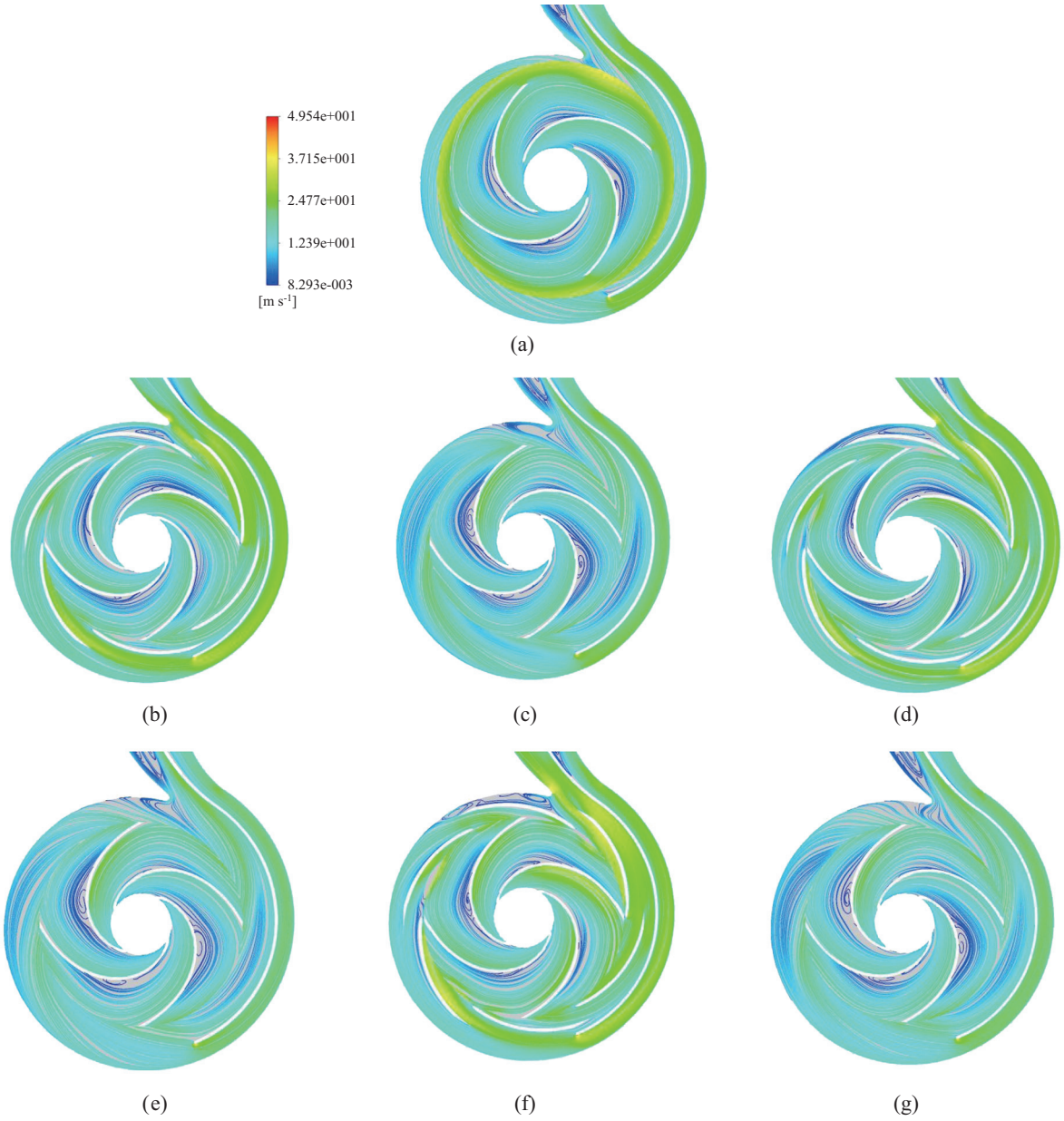


Fig. 10 Stream function (a) impeller without diffuser (b) impeller with vaned diffuser α = 0° (c) impeller with non-vaned diffuser α = 0° (d) impeller with vaned diffuser α = 5° (e) impeller with non- vaned diffuser α = 5° (f) impeller with vaned diffuser α = 10° (g) impeller with non- vaned diffuser α = 10°.

Velocity magnitude contours of different cases have been illustrated in Fig. 11 at mid- span of the impeller. Figure 11(a) shows less uniformity for velocity field of impeller without diffuser compared to other cases equipped with impeller diffuser. By installing the impeller diffuser on the downstream of the impeller the uniformity has been increased due to the beneficial influences of impeller diffuser. Indeed, more uniform distribution of velocity field for impellers with vaned diffuser (see Figs. 10(b) and (d) and (f)) which resulted in higher performance is due to contribution of impeller diffuser in conversion of kinetic energy to potential one.

In Fig. 12 stream function at volute cross section of all cases have been shown. According to the figure, secondary flows for the impeller with no diffuser (Fig. 12(a)) are asymmetry as well as cases with non vaned diffusers (Figs. 12(c), (e) and (g)). In fact, one of the secondary flows becomes stronger in the pointed out figures with increasing wall diffuser angle (α) as the strongest one is seen in the case of non- vaned diffuser α = 10°. Whereas, symmetry secondary flows are obtained for the cases of vaned diffuser α = 0, 5 and 10° (Figs. 12(b), (d) and (f)) thanks to the role of diffuser vanes. Moreover, the strength of secondary flows reduces with enhancing wall diffuser angle (α) of vaned diffusers as they are dying out in the maximum investigated α = 10° (Fig. 12(f)) and more uniform flow is accomplished in this case.

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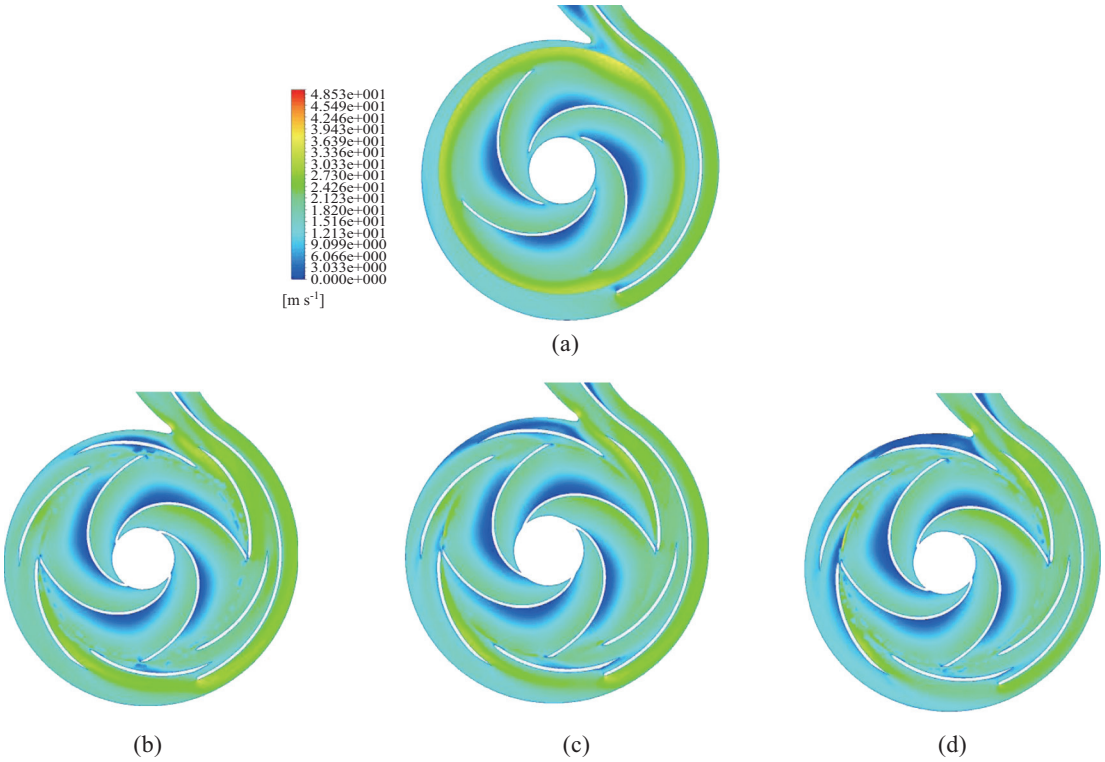


Fig. 11 Velocity magnitude contours (a) impeller without diffuser (b) impeller with vaned diffuser $\alpha = 0°$ (c) impeller with vaned diffuser $\alpha = 5°$ (d) impeller with vaned diffuser $\alpha = 10°$.

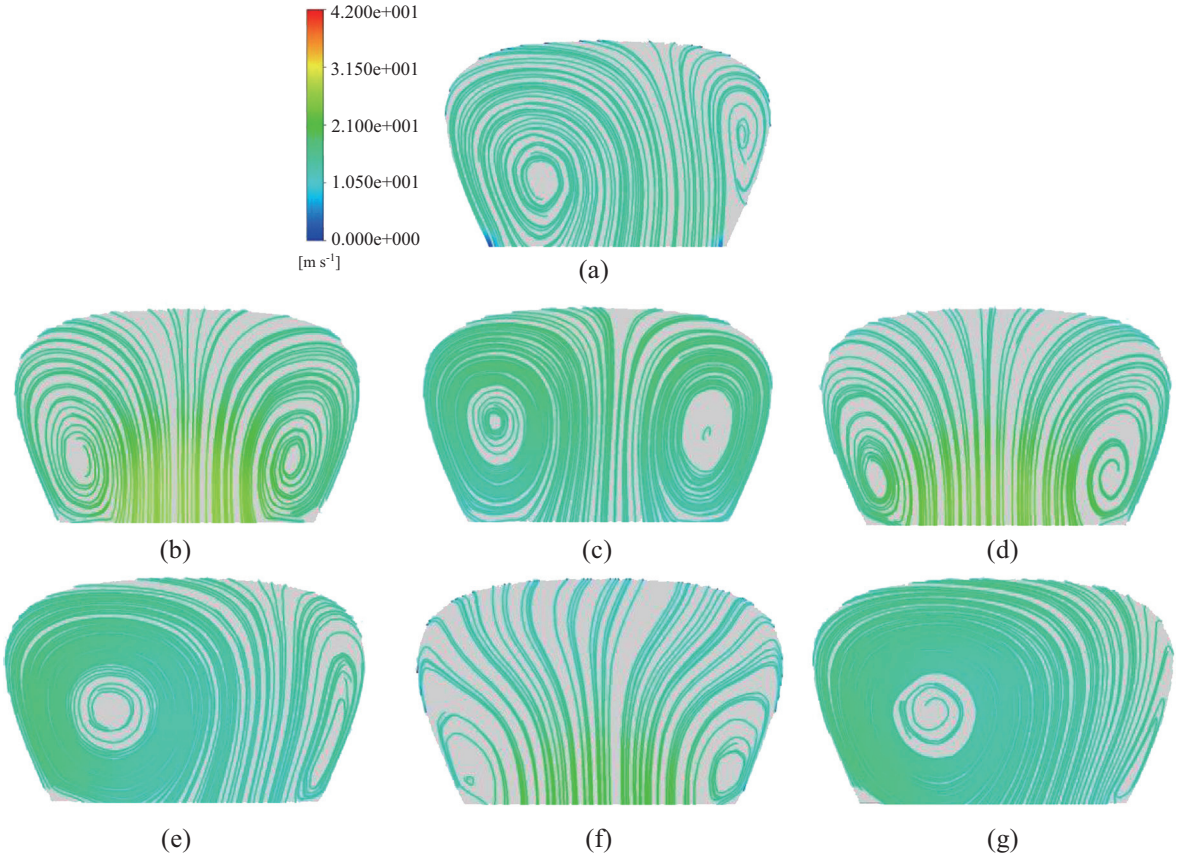


Fig. 12 Stream function at volute cross section (a) impeller without diffuser (b) impeller with vaned diffuser $\alpha = 0°$ (c) impeller with non- vaned diffuser $\alpha = 0°$ (d) impeller with vaned diffuser $\alpha = 5°$ (e) impeller with non- vaned diffuser $\alpha = 5°$ (f) impeller with vaned diffuser $\alpha = 10°$ (g) impeller with non- vaned diffuser $\alpha = 10°$.

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6. CONCLUSIONS

The following conclusions are achieved based on the computational and experimental study on the influences of diverging angle of impeller diffuser α, with and without vane, ranging from 0° to 10° on the performance characteristics of a centrifugal pump.

1. The change in divergence angle (α) of the diffusers has significant effect on the head and overall efficiency of the centrifugal pump. The influence is remarkable at all flow rates particularly at high flow rates.
2. The highest head and efficiency at high flow rates are obtained for the angular vaned diffuser of α = 10°, while at low flow rates they are achieved by vaned diffuser of α = 0°. Furthermore, the lowest head and overall efficiency is belonged to the centrifugal pump without vaned diffuser. The highest head at BEP is obtained when α = 10°, and it is 12.3% compared to that of α = 0° and at high flow rate that is 1.15 times higher than that for impeller without diffuser.
3. Among all investigated cases for diffusers, the vaned diffuser of α = 10° had the best effect on the overall efficiency and it enhanced by 9 and 2.7 percent with respect to the centrifugal pump without diffuser at high flow rate and BEP respectively. Moreover broadened operating region for the centrifugal pump was accomplished by using vaned diffuser α = 10° as well.
4. Results from this study inspire more investigation on diffusers angles for centrifugal pump impeller as they can be employed to solve problem of restricted operating region of centrifugal pumps and they can also increase overall efficiency of centrifugal pumps.

The highly important point is that due to the restriction of casing volute diameter in one hand and a need for sharply enhancing the diameter ratio of a conventional impeller diffusers (α = 0°) for achieving higher pressure recovery in other hand is the most challengeable problem regarding the larger conventional impeller diffusers that could cause higher friction losses. Tackling such an issue by utilizing angular impeller diffusers (α > 0°) is possible as revealed in the present investigation that by making main diffuser walls divergent higher centrifugal pump performance is achieved with maintaining same diameter ratio of an impeller diffuser.

NOMENCLATURE

F Source term;

H Head;

N RPM;

P Pressure;

Q Flow Rate;

T Stress tensor scale;

U Velocity;

Flow coefficient; $\frac{Q}{nD^3}$

Head coefficient; $\frac{gH}{n^2D^2}$

Reynolds number; $\frac{\rho nD^2}{\mu}$

Specific Speed; $\frac{NQ^{0.5}}{H^{0.75}}$

α Divergence angle of the main body walls of impeller diffuser;

ρ Density;

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