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Numerical and Experimental Study of a Centrifugal Pump with Varying the Outlet Short Blade Angle

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Abstract: *Around 20% of the world total energy consumed by pumps. Centrifugal pump is the most common type used in all industrial and agriculture application. The challenge is how to save the energy consumption due to pump work. Slip factor is the most important parameter which effect on the efficiency of the centrifugal pump. In the present paper, the performance of a centrifugal pump with specific speed 46 is studied numerically and experimentally and the slip factor is predicted numerically. The effect of adding short blades with varying the outlet blade angle of the shorted blades also is studied experimentally and numerically. The numerical result of two cases of the impeller, with and without short blades, has a good agreement with the numerical one. Adding short blade has a good effect on the slip factor magnitude which improved by 21 %. Also the total efficiency of the centrifugal pump increase by 1.9 %.*

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Nomenclature			
σ	Slip factor	Ns	Specific speed
δ	Deviation angle	Q	The pump discharge m^3/hr
β_{2B}	The ideal exit vane (flow) angle	H	The pump head m
β_2	The actual exit vane (flow) angle	Ω	The pump speed RPM
Z	The number of impeller's vane	G_k	generation of turbulent kinetic energy
u_r	vector fluid velocity in the rotating system	$\sigma_k,\sigma_\varepsilon$	turbulent Prandtl numbers
μ_{eff}	dynamic effective viscosity	L	Cord length of the main blade
u	vector fluid velocity in the stationary frame	L_R	Cord length of the short blade
k	turbulent kinetic energy	Q_n	Nominal pump discharge m^3/hr
ε	dissipation rate	d_4	Volute out diameter mm
μ	laminar viscosity	b_3	Volute thickness mm
μ_t	Turbulent viscosity	t_{blad}	Blade thickness mm
Φ	impeller discharge flow coefficient, dimensionless C_{m2}/U	ρ	Density of fluid kg/m^3

I. Introduction

The performance of pump is estimated by the dynamic of the fluid inside the pumps getting from the designing structure. The rotational element of the pump is called impeller which the most important part that effect greatly on the overall efficiency of the pump performance. The total head for the centrifugal pump is the resultant which can be estimated as the developed theoretical head, which the minus result between Euler's equation, and hydraulic losses. The impeller with small number of blades does not guide the fluid flow properly and an additional mixing loss is occurred. When the blades number increase, the slip factor increase and the theoretical head of the pump increase, while too many numbers of blades get decreasing in the efficiency according to the increasing of blockage at inlet and the friction through the impeller passage. A method, which considers a type for modifications of increase the slip factor, theoretical head and also the efficiency developed by the impeller is using shorted blades that reduce the circulation near the impeller exit. The short blade is partial blade with a shorter in longitudinal length than the original blade and casted in the space between each two original blades.

The characteristic curves of the impeller with and without splitter blades were studied experimentally to show the effects of lengths and the number of blades of the splitter blades on a deep well pump performance [1]. Positive splitter blade effects are not seen for the high number of blades and low blade discharge angle. However, adding splitter blades is useful for the impellers with small number of blades (z = 3 and 4). When the splitter blades with a length of 80% of the main blade length were added to the same blade number, the energy consumed by the deep well pump decreased.

Abdrabo et al. [2] have studying a characteristic of centrifugal pump with shorted blades experimentally and were found using shorted blade of length 24% from full length blade improve efficiency and head of the pump by 8% and 9% respectively.

Liuhoulin et al. [3] with increases number of blades impeller the uniformity distribution of vapor in impeller channel became obvious. The head of centrifugal pump grows all time, the change regulation of efficiency and NPSH are complex, and the area of flow pressure region at the suction of blade inlet grows continuously.

Guang li [4] a large exit blade angle results in an increase in the hydraulic loss over the entire flow rate range in the volute. However, it caused an increase in hydraulic loss at a low flow rate and reduced the loss at a high flow rate in the impeller. The flow in the impeller with a large exit blade angle acquires a separated structure near the pressure side of the blade even at the point of best efficiency. Elucidating the evolution of such a flow pattern in the impeller by means of more advanced turbulence flow models at various working conditions is a challenge for hydrodynamics researchers.

Massinissa et al. [5] get the better performance for design parameters of the centrifugal pump, The results show improving in the pump head and the brake horsepower according to increasing the impeller blade number and impeller blade height. The designs parameters which is selected also effect on pump overall efficiency.

Rababa [6] the efficiency for the simulation model of the pump tap with two-, three- and four-bladed impeller wasn't change. The experimental estimated values of efficiencies for the pump have a good agreement with simulation results. The pressure head is not influenced by the length of the additional shorted blades, which are added between the main blades. The pump head increases by 4% according to the presence of short blades, with significantly reduced flow areas in the wheel.

Sun-Sheng et al. [7] the comparison of numerical predicted performance for impeller with and without splitter blades presents that the efficiency increased and pressure head, which is required for that efficiency, is decreased when splitter blades are added to impeller flow passage.

Guang LI [8] studded the blade exit angle effects on the standard industrial Centrifugal Oil Pump performance. The blade exit angle of the blade has a great but same influence on the pump head, shaft power and efficiency at various viscosities of liquids, which pumped to the centrifugal oil pump. The degradation of the pump performance with increasing viscosity of liquids caused by a rapid reduction in the hydraulic and mechanical efficiencies.

Abdrabo et al. [9] mid-way shorted blade at impeller exit improve the efficiency along pump performance and caused a 30% increase at the operating point, which compared with the original one at 2900 rpm. The impeller has a splitter blade near to the pressure side got modification in pump efficiency at all speed, which compared with the standard impeller, but lower than the improvement of mid-way shorted blade impeller.

Said. A. F. Hawash et al [10] evaluate the slip factor value with and without adding short blades from the results of pump simulation using CFD technique. In this research the value of slip factor of the stander impeller has a good contacts with the value of slip factor estimated by stanitz formula.

Abdullah H. I. Aboelnil et al [11] predict the slip factor value for radial and mixed flow pump. Numerical work of sixteen radial and mixed flow pumps have been presented in this paper aimed to predict the slip factor of each case using new technique. The relationship of slip factor and specific speed is not affected unless

adding the effect of the flow coefficient and the flow exit velocity.

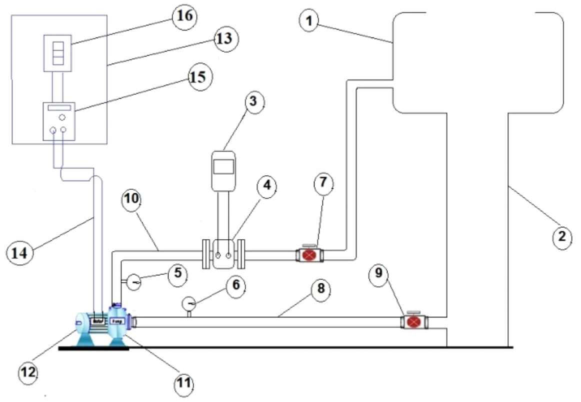
The dereliction of the previous studies is studying the relationship between the slip factor and the outlet vane angle of the impeller especially for splitter blades.

In the present work the effect of adding short blades on the slip factor is studied with varying the outlet blade angle of shorted blades to show the relation between the outlet blade angle of shorted blade and the slip factor which presented in [11] where concluded that, there is inverse fit between the slip factor value and the impeller exit blade angle, the modification effect on the performance of the centrifugal pump with numerical and experimental study.

II. Experimental work

The experimental test rig and measurement instruments are presented schematically in figure 1. The experimental set up contains the electrical motor, which derive the pump has power of 5.5 kW and 2900 with water as a working fluid in a closed loop. The electrical motor is connected directly to variable frequency drive to change the pump speed. The pump has round cross section volute casing and one stage in impellers. The Pump has a suction and discharge nozzles with diameter 90 mm. and also has aback word and closed type impeller. For the discharge monitoring a magnetic flow meter with accuracy 0.01 m^3/hr is used. Pressure at the pump deliver and suction, which below atmospheric pressure, are monitoring by two calibrated pressure gauge.

The suction gauge pressure has range from -1 to 1 bar and accuracy 0.05 bar while the discharge one with range from 0 to 7 bar and accuracy 0.1 bar.



1	Open tank	9	3 inch gate valve
2	12 inch pipe	10	3 inch discharge pipe
3	Flowmeter display unit	11	Pump
4	3 inch magnitic flowmeter	12	Motor Drive
5	Discharge pressure gauge	13	Elictrical Board
6	Suction pressure gauge	14	Elictrical Wire
7	3 inch ball valve	15	Variable speed drive
8	3 inch suction pipe	16	Elictrical conductor

3a

3b

Fig 2 Experimental test rig (a) scheme diagrams and (b) Photograph of the test rig

III. Impeller Modification

The centrifugal pump which used in the present work with specific speed 46 EBARA model and the main dimension of the pump are shown in the table(1)

Table 1 dimension for tested pump

No of blades	6	Impeller blade thickness	2.3 mm
Shaft diameter	18 mm	Impeller outlet angle	21 mm
Suction diameter	90 mm	Volute inlet diameter	150 mm
Impeller diameter	135 mm	Volute inlet width	30.7 mm
Outlet width	20 mm	Volute outlet diameter	80 mm

The pump performance is monitored with the five impeller configuration cases A to E. All cases have the same in slandered dimension as the original impeller A. The configurator impellers from B to E are modified impellers with shorted blades. the variation of shorted blade outlet angle from B to D is shown in table (2)

Table 2 the configuration impeller dimension

	Short blade cord length (L_R/ L)	Outlet blade angle for main blades (deg)	Outlet blade angle for short blades (deg)
A	No short blades	21	No short blades
B	16 %	21	21
C	16 %	21	20
D	16 %	21	19
E	16 %	21	18

The choosing length of shorted blade 16 % from the main blade full length is the best case of adding splitter blades for this centrifugal pump according reference [12]. The variation of outlet blade angle for shorted blade cases is shown in figure 2 and the geometry of all impeller cases has shown in figure 3.

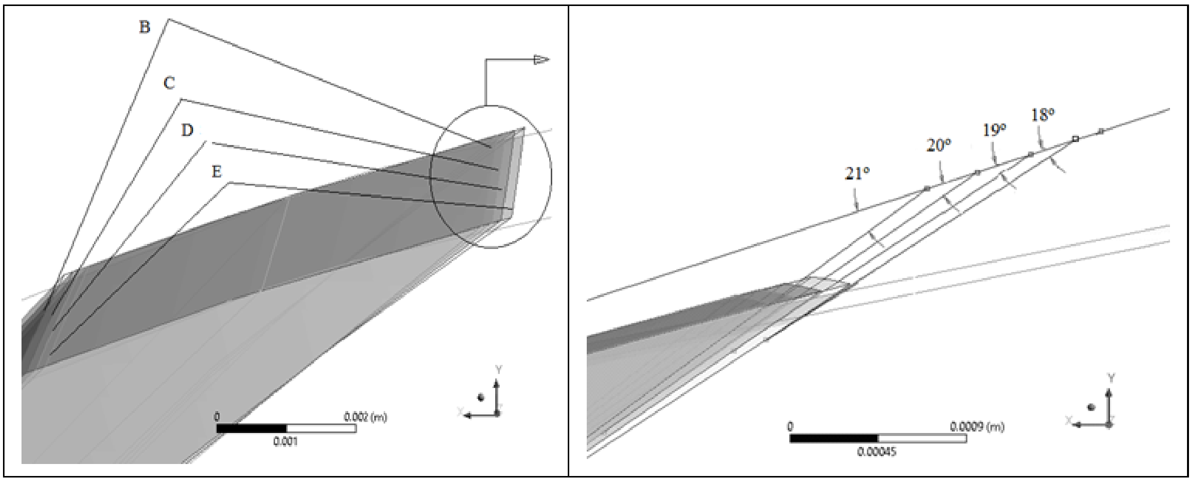
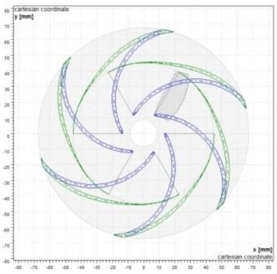
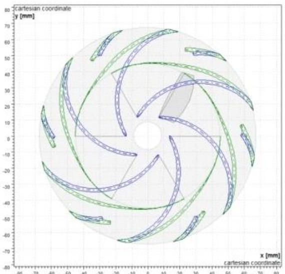


Fig 2 Variation of outlet blade angle



A



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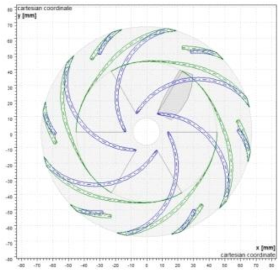

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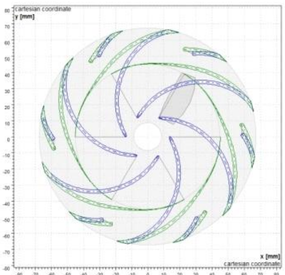

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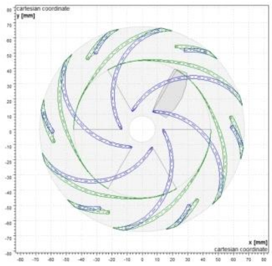


E

Fig 3 geometry of all impeller cases

IV. Numerical Model

The centrifugal pump with a complete 3D model is design by a turbo machine program. The pump with closed type impeller and circular cross section volute casing as shown in figure 4 all impellers, the original and configuration, are designed by the same dimension listed in table 1, 2. The suction and the deliver pipe are designed with length fifth of their diameter to standardize monitor the suction and deliver pressure.

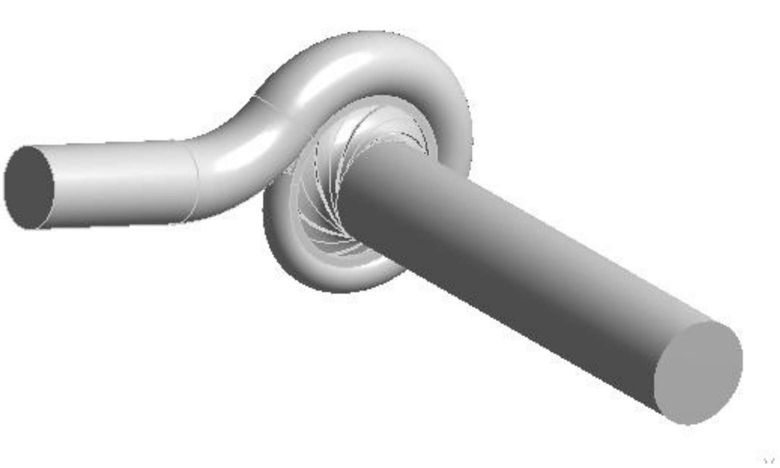


Fig 4 complete 3D model for the simulated pump

V. Grid Generation And Boundary Layers

The better grid generation of the simulation pump model gets the better description of the results for the flow domain around the model. All small spaces between the pump casing and the impeller shroud are assumed neglected in the numerical simulation. The 3D simulation volume has three zones. The first, inlet pipe, and third zone, the volute casing, are not rotationally zone while the second zone that consist the moving blades by rotational speed of Ω=2900 rpm.

The suction or inlet pipe is represented as the first zone represents that is 90 mm in diameter while the third zone represent the volute casing and discharge pipe. The second zone consists of the hub, shroud and the moving vane of the pump. The three zones were connected by interfaces boundary condition.

Fluent's pre-processor generate the mesh of the geometry and the computational pump domain. Unstructured mesh type is refine to present the inlet and outlet boundary condition. Also tetrahedral mesh is generated for the domain of impeller and volute casing as shown in Fig 5.

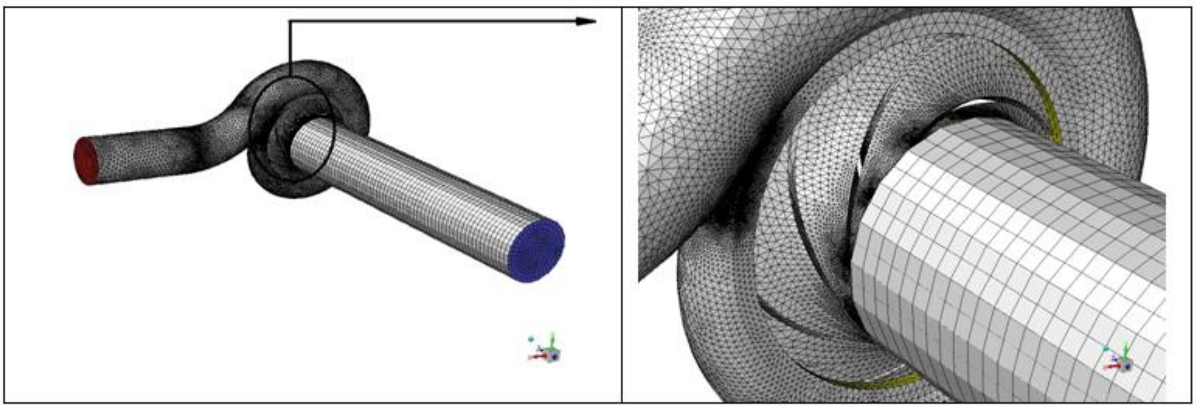


Fig 5 Sketch of the unstructured mesh of the pump.

Complete 3D tetrahedral cells mesh file is generated with 550000 nodes. The inlet and the outlet boundary condition of the pump model are identified as mass flow inlet and mass flow outlet. The contact faces between the moving and stationary zone are represented as interface boundary condition. Also the stationary and the moving walls are identified with respect to their zone.

VI. Governing Equation And Solver

For the rotating impeller the incompressible flow is solved in a reference moving frame with value of same rotational speed of the vane speed. For the stationary parts the flow is solved as inertial reference frame. The governing equations for the rotating vane are shown in continuity and momentum equation 1, 2 respectively [13].

$$\nabla \rho \mathbf{u}_r = 0 \ldots \qquad (1)$$

$$\nabla \rho \mathbf{u}_r + 2\rho \mathbf{\Omega} \times \mathbf{u}_r + \rho \mathbf{\Omega} \times \mathbf{\Omega} \times \mathbf{r} = -\nabla p + \mu_{eff} \nabla^2 \mathbf{u}_r \ldots \qquad (2)$$

While μ_{eff} is the dynamic effective viscosity. In the left hand side of equation (2), the last two terms are defined as the effects of the centrifugal and Coriolis forces according to the reference of rotating frame.

$$\nabla \rho \mathbf{u} = -\nabla p + \mu_{eff} \nabla^2 \mathbf{u} \ldots \qquad (3)$$

The governing equations for the fixed parts of the modeling zones are solved as stationary reference frame. So the equation of continuity remains constant, but the momentum equation changed to equation 3.

The standard turbulence k-ε model solves the flow [13], because of robustness and reasonable accuracy. The differential transport equations for the turbulence kinetic energy and turbulence dissipation rate are shown in equation 4, 5 respectively.

$$\nabla \rho \mathbf{u} k = \nabla\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\nabla k\right) + G_k - \rho\varepsilon \ldots \qquad (4)$$

$$\nabla \rho \mathbf{u} \varepsilon = \nabla\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\nabla \varepsilon\right) + C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \ldots \qquad (5)$$

$$\mu_t = \rho C_m \frac{k^2}{\varepsilon} \ldots \qquad (6)$$

Which values of the constant for the model are $C_{1\varepsilon}$=1.44, $C_{2\varepsilon}$=1.92 and C_m=0.09 according reference [13].
The calculations of the present model have been solved by Fluent CFD that uses the finite volume solving type for the steady 3D incompressible Navier-Stokes equations solution. The selection turbulence model is the standard k-ε model. The involved parameters which are the turbulence intensity and the hydraulic diameter are estimated with values of 5% and D/2, respectively. The SIMPLE algorithm is used for the pressure-velocity coupling. Convections terms and central difference schemes for diffusion terms are performed through second order, upwind discretization.
The FLUENT program is solved in a 3.4GHz Pentium IIV PC. The scaled residual is determined by the value of 10^{-5}, as criterion of convergence.

VII. Model Validation, Results, And Discussion

A complete test of the pump model, for case A, is simulated numerically with different rotational speed 2900, 2500, 2000, and 1400 RPM as shown in figure 6 each case of rotational speed is tested with different values of discharge inter the inlet pipe of the model such as in the case of 2900 RPM as shown in figure 7 the model is simulated with Q/Q_n values equal 0.74, 0.85, 1, and 1.08. Under the assumption of no volumetric leakage the pump model approximation and the experimental measured uncertainty the 10% difference between the experimental head and the CFD head is accepted according reference [14], in the present model for all tested point the difference between CFD results and the experimental results not excised this value of uncertainty.

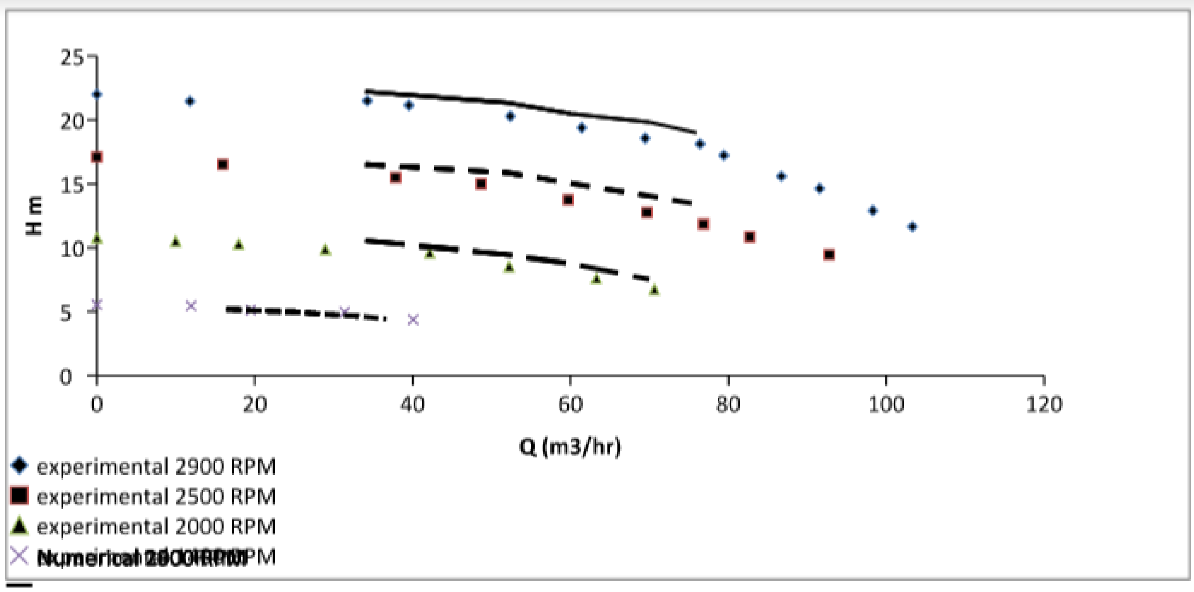
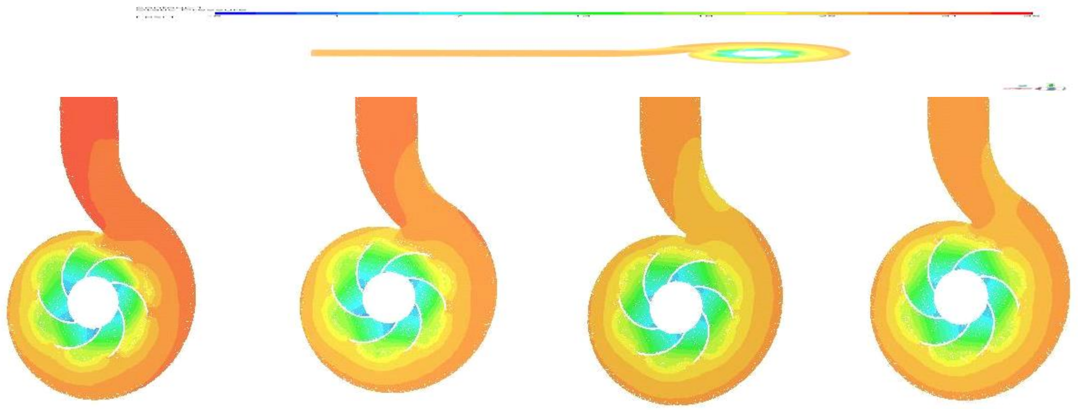


Fig 6 experimental and numerical results of tested pump for case A



Static press contour Q/Q_n 0.74 | Static press contour Q/Q_n 0.85 | Static press contour Q/Q_n 1 | Static press contour Q/Q_n 1.08

Fig 7 Numerical results of tested pump for case A at 2900 RPM

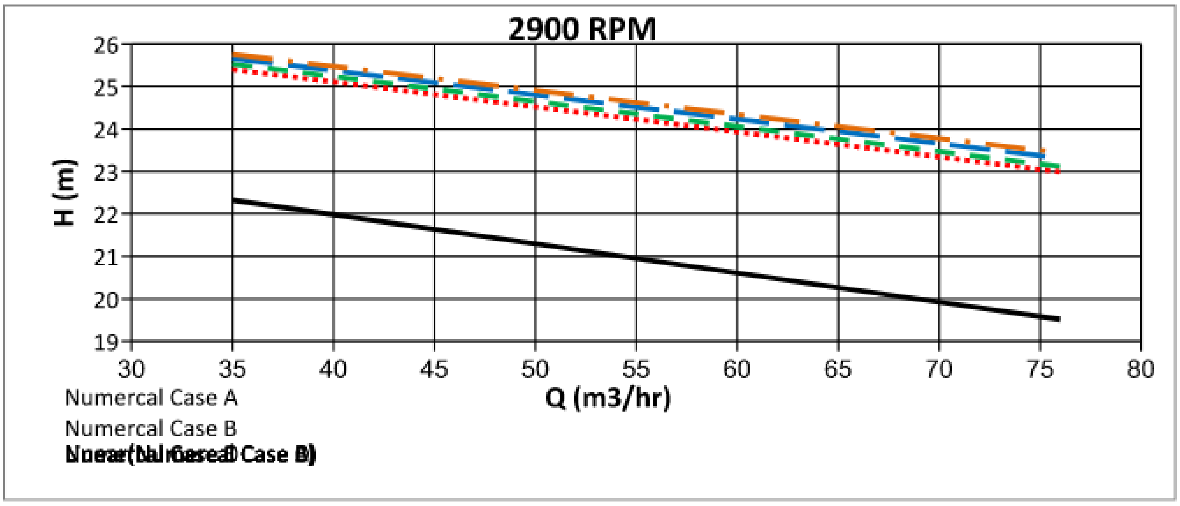
Each five cases of impeller configuration are completely simulated at different rotational speed and at each rotational speed has a several cases of discharge inlet the suction pipe. Theoretical head of the pump model is predicted from monitoring the suction, the discharge pressure and velocity, according Bernoli equation, for all tested point. According to reference [11] which slip factor prediction equation as shown in equation 7

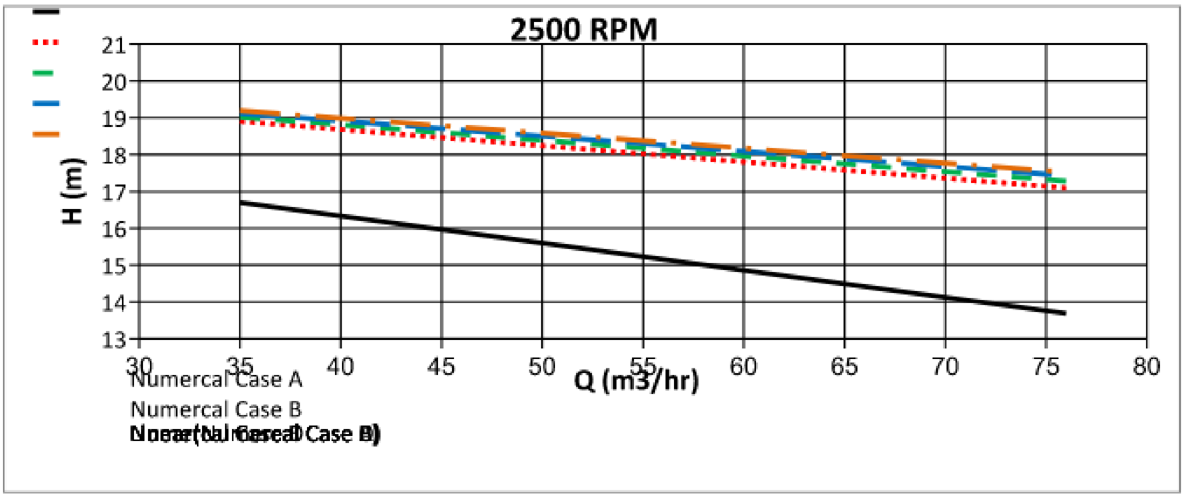
(7)

Equation (7) enables the effect of the three design parameters Φ, N_s and β_2 on the slip factor and so to the total head. If Φ is reduced, then the total head is increased, while the value of the vane angle is inversely proportional to the slip factor value.

In the present study at all rotational speed, the total head is improved with adding short blades to the main blades of the impeller as shown in figure 8 For adding short blades with angle β_2 =21º the total head increases by 15 % at the nominal flow and 2900 RPM the effect of change the vane discharge angle is clearly shown at all rotational speed cases. The total head improved with 16%, 17.5%, and 17.9% when the β_2 changed to 20º, 19º, and 18º respectively at the nominal flow and 2900 RPM.

The total head is estimated from the static pressure values at the suction and discharge face of the model. The static pressure at the outlet face of the volute case is increases with adding splitter blades to the main blades of impeller also the change of van outlet angle of the short blades has a good effect to this values as shown in figure 9 which the value of static pressure at the outlet face of the volute case for case A, B, C, D, and E are 1.97, 2.27, 2.28, 2.31, and 2.32 $*10^5$ pascal respectively.





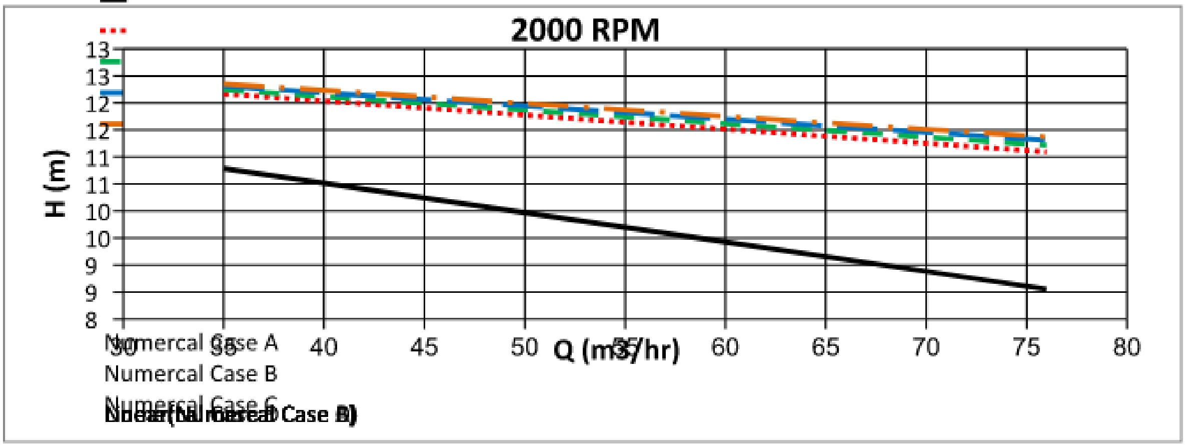
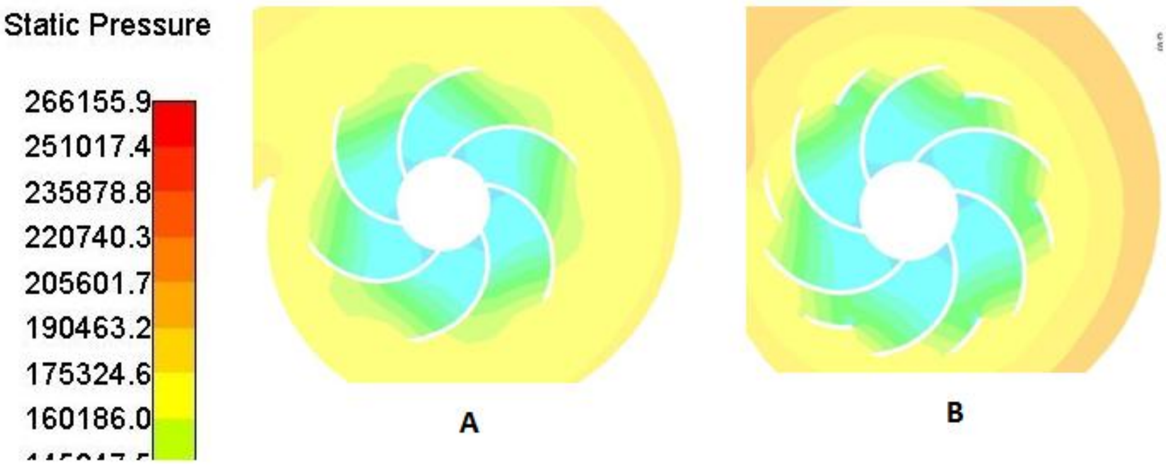


Fig 8 numerical results for all cases



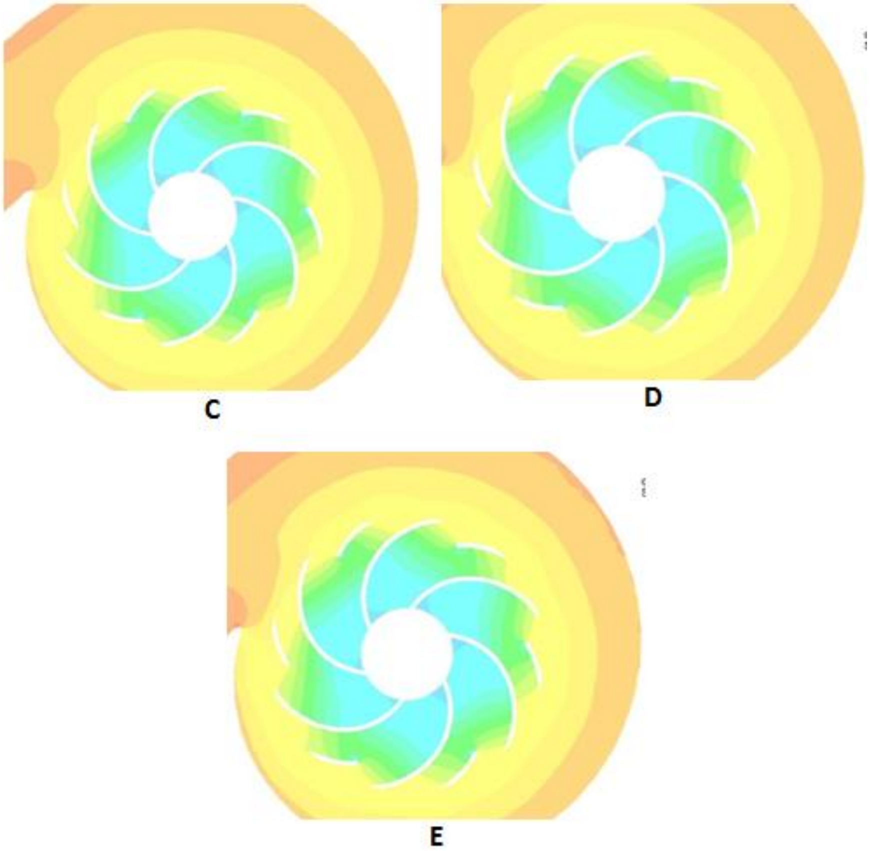


Fig 9 Static pressure contour

The improvement of the static pressure with adding short blades due to the increasing of the velocity magnitude leaves the impeller according the effect of increase the number of compact blade faces which transfer the energy to the fluid particles.

The effect of adding short blades is track the calendar of the fluid path line exit the impeller face to reduce the effect of slip occurred between the impeller main blades by correcting the outlet angle of the fluid leaves the impeller.

In the present study, due to the correction of the fluid outlet angle exit the impeller by adding short blades, the velocity magnitude contour shows the improvement of the velocity distribution of spam between the blades which in case A, without short blades, has a bad velocity distribution between each two blades, the velocity values increases at the suction and the discharge side of the blade while decreases at the middle area between each two blades, as shown in figure 11 this is the phenomena of slip between the blades of the impeller. The splitter blades improve the slip factor of the impeller due to increase the value of the velocity of fluid leaves the impeller. At the present model the average values of the velocity magnitude at the impeller exit face of all cases is estimated from the histogram of the velocity magnitude shown in figure 10 For case A the average velocity value is 11.2 m/s the effect of adding short blades for the impeller of case A is clearly shown when estimate the average value of the velocity magnitude for case B which equal 12.3 m/s. also the effect of changing the outlet vane angle of the splitter blade for the present simulated pump is shown when estimate the values of the velocity magnitude for cases C, D, and E which equal 12.6, 12.8, and 13 m/s respectively.

According to reference [10]and [11] which estimate the slip factor from plot the ideal exit triangle and the actual exit triangle using the estimated absolute velocity magnitude as actual absolute velocity leaves the impeller, the values of slip factor of modification impellers improved from 0.7 For case A to0.82, 0.83, 0.85, and 0.86 For case B, C, D, and E respectively.

The effect of increasing slip factor of the impeller is also shown in figure 12 which shows the correction of the fluid path line exit the impeller face, the vector show the different between the angle of the vector leave the impeller for all cases. For case A the vector shows the variation of the fluid vector exit angle from the suction and discharge side of the blade, while after using short blades which modify the vector path angle across the middle spam area between the blades, this modification increase with decrease the outlet vane angle of the short blade as shown in figure 12.

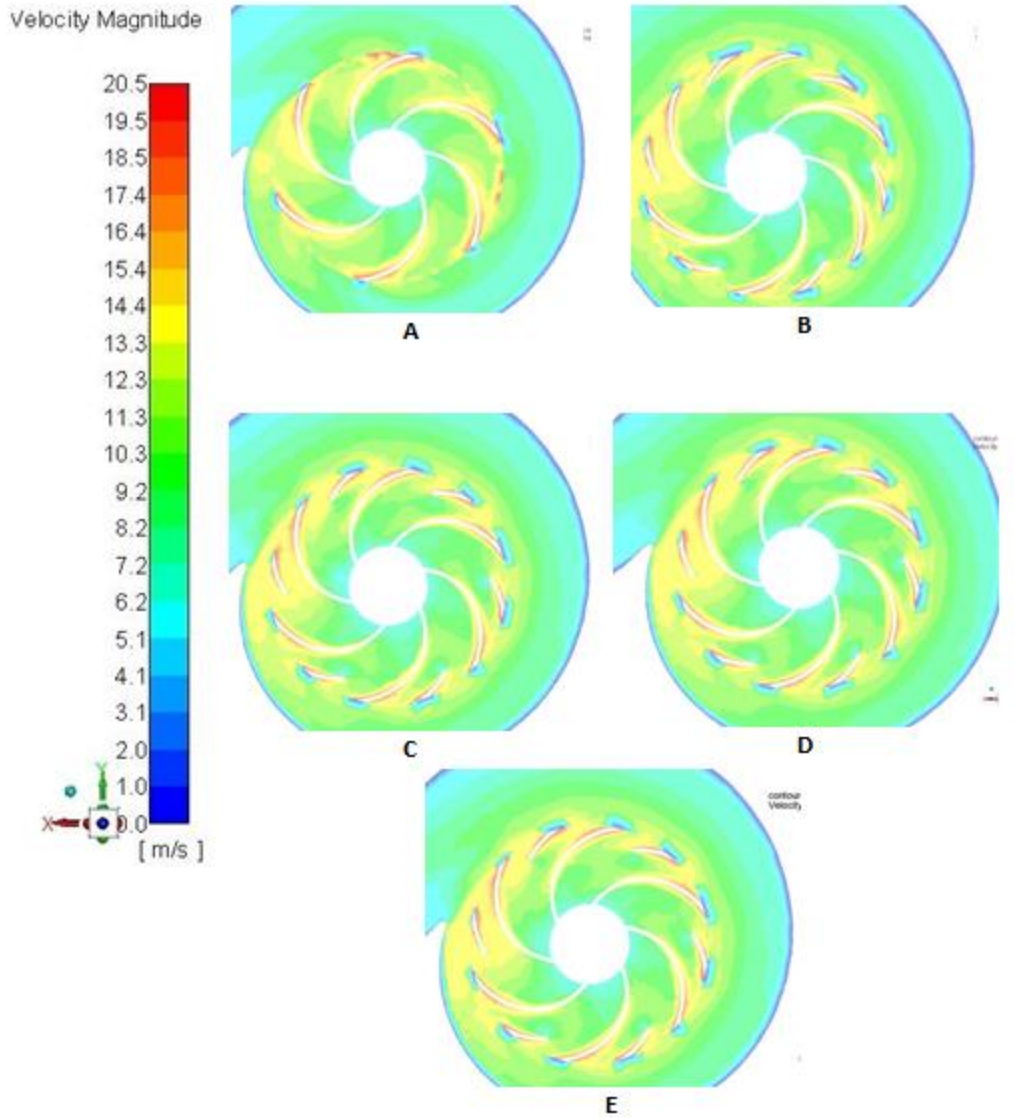
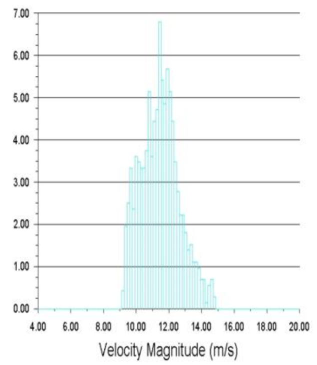
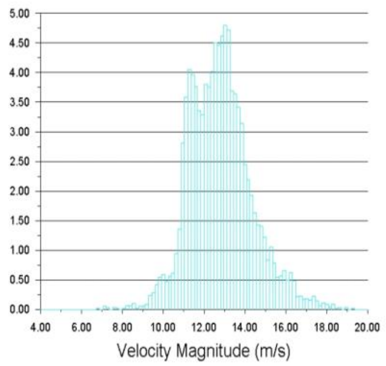


Fig 10 Absolute velocity magnitude contour



A



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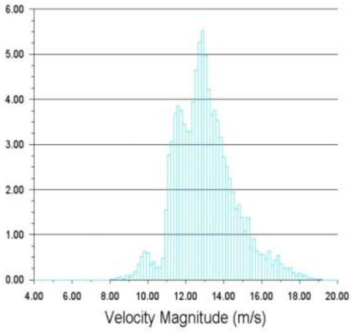

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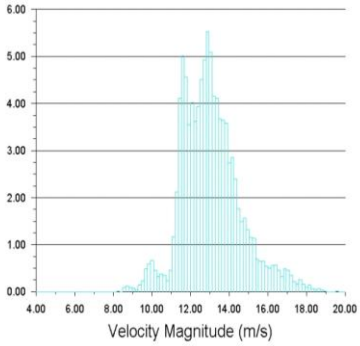
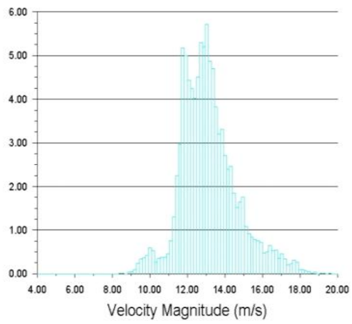


D



E

Fig 11 Histograms of the absolute velocity values leave the impeller

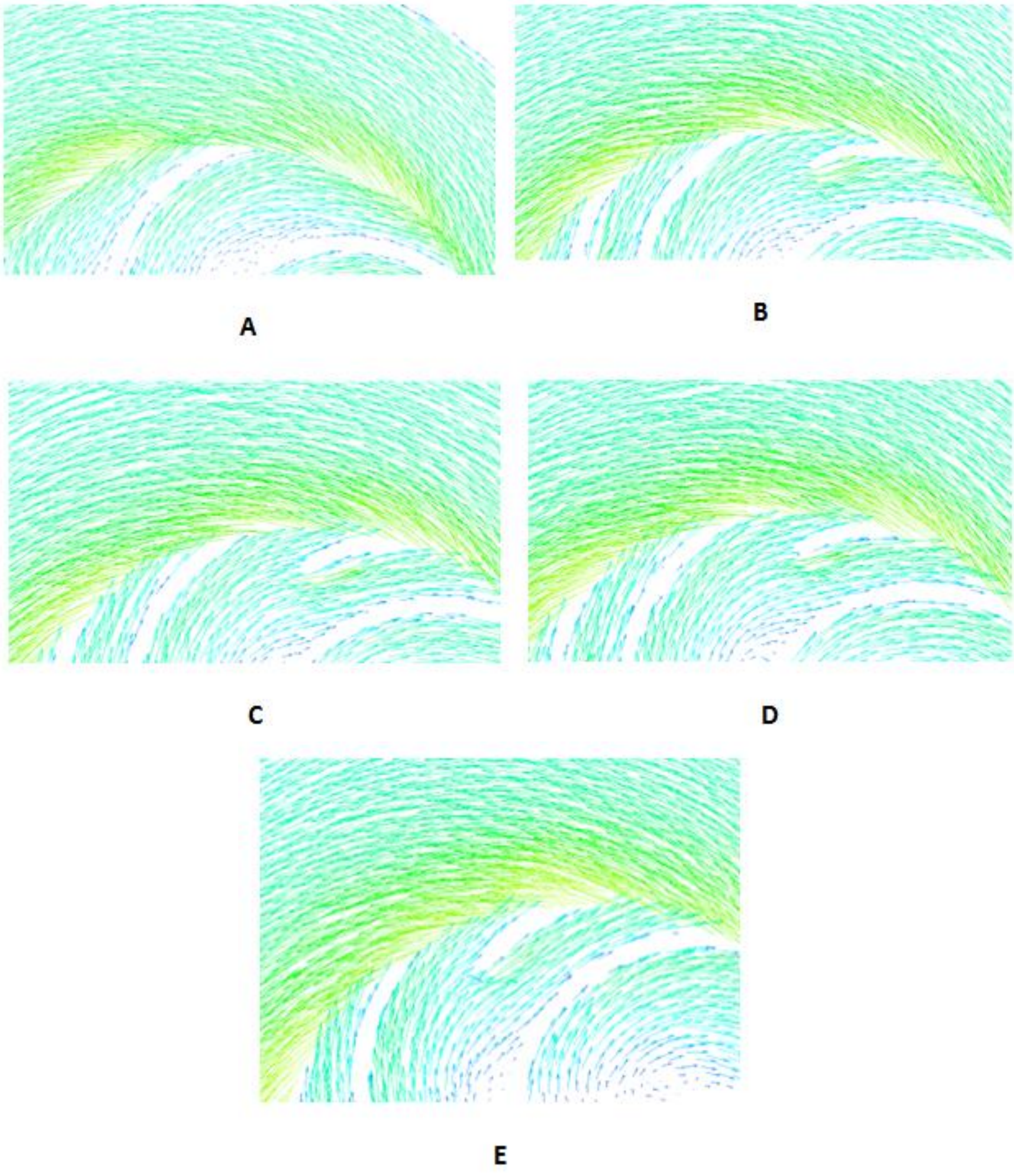


Fig 12 Vectors of flow exit the impeller

VIII. Experimental results

A Complete test is done for the present centrifugal pump used in this study. Flow, suction and discharge pressure, input power, and rotational speed are monitored for all tested point using calibrated instrument. Five impellers are casted and installed to the pump for test. The pump is tested with each five impellers and each case tested at three different rotational speed 2900, 2500, and 2000 RPM.

For all cases of impeller modification three curves are derived, flow against head, flow against input power, and flow against total efficiency curve.

The short blade effect is shown in figure 12 which shows the experimental results of tested pump for all cases that improve the total head by 1.6%, 2.7%, 3.2%, and 3.2% for case B, C, D, and E respectively at the nominal flow and 2900 RPM.

The total efficiency of tested pump also improved with the modification of short blades which increase by 1.3%, 1.7%, 1.9%, and 1.9% for case B, C, D, and E respectively at the nominal flow and 2900 RPM.

Figure 13 and 14 shows the experimental results at 2500 and 2000 RPM respectively. The results of each rotational speed give improvement at the performance and efficiency for the modification cases

There is a shift between the numerical and experimental improvement percentage. The reason which effect and show this shift is the Pfleiderer design type used in the present solve does not take care of the complicated 3-D flow structures and the empirical formulas are applied for the account of hydraulic losses, as represented in reference [13]. Also the difficulty of casting the modification impeller with small varying of the outlet vane angle with exact finishing and real design angles dimension compare with the simulation impeller model which design by 3-D catia programs is may be considered.

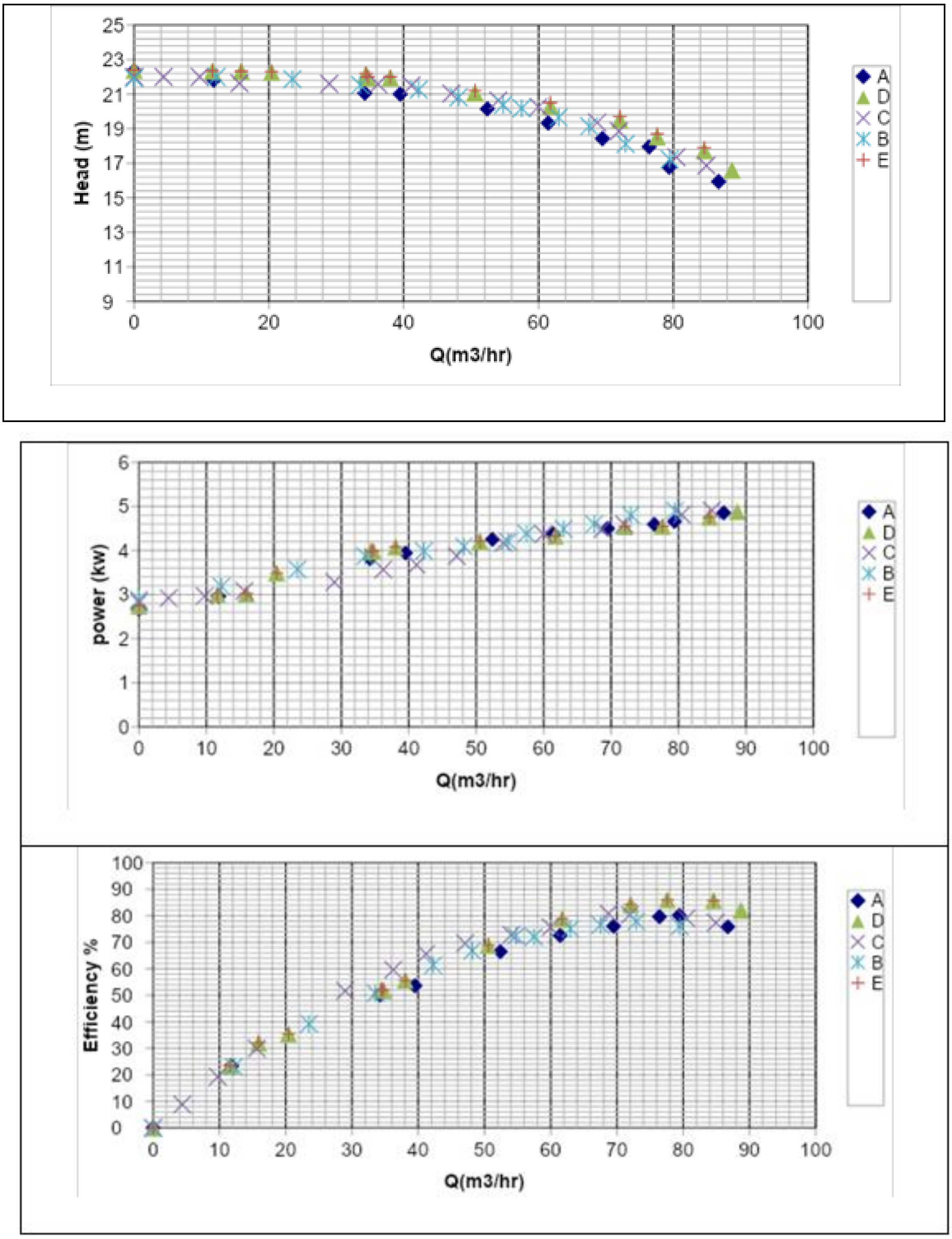
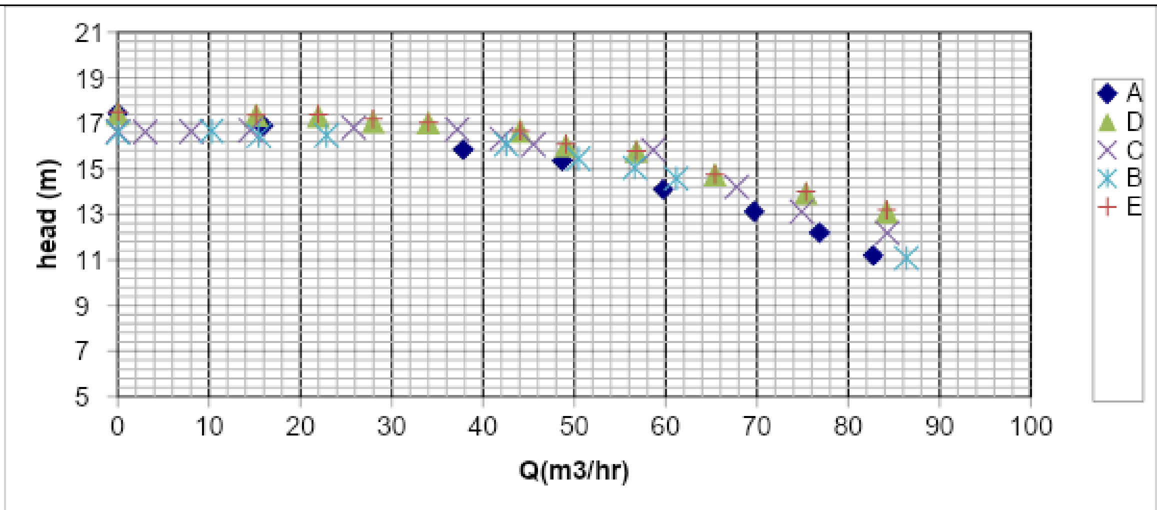
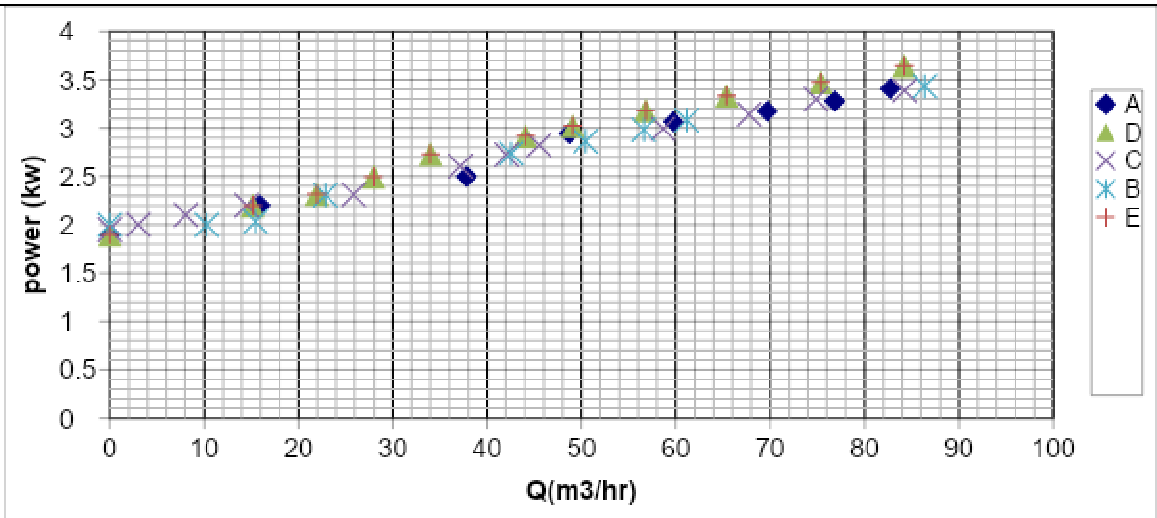


Fig 12 experimental results at 2900 RPM





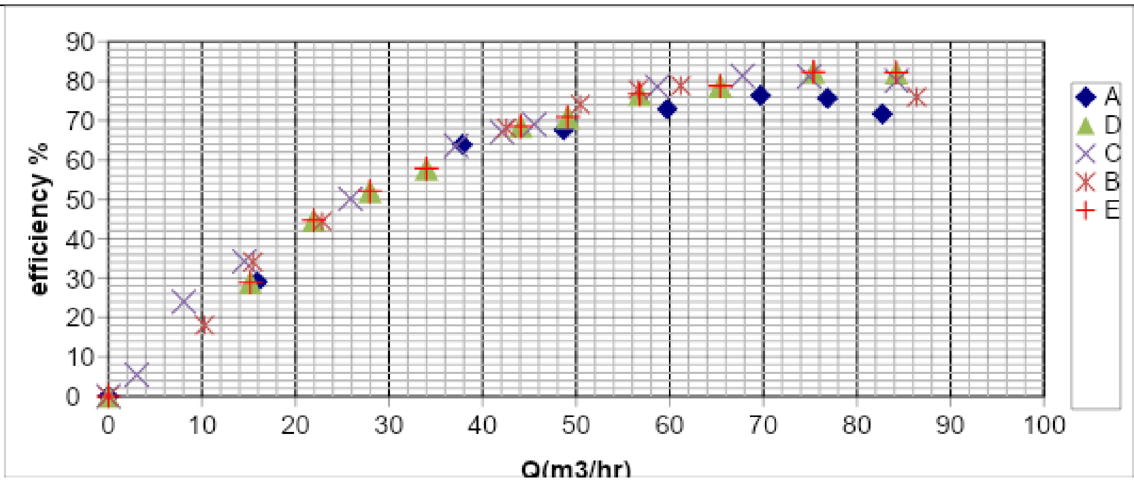


Fig 13 experimental results at 2500 RPM

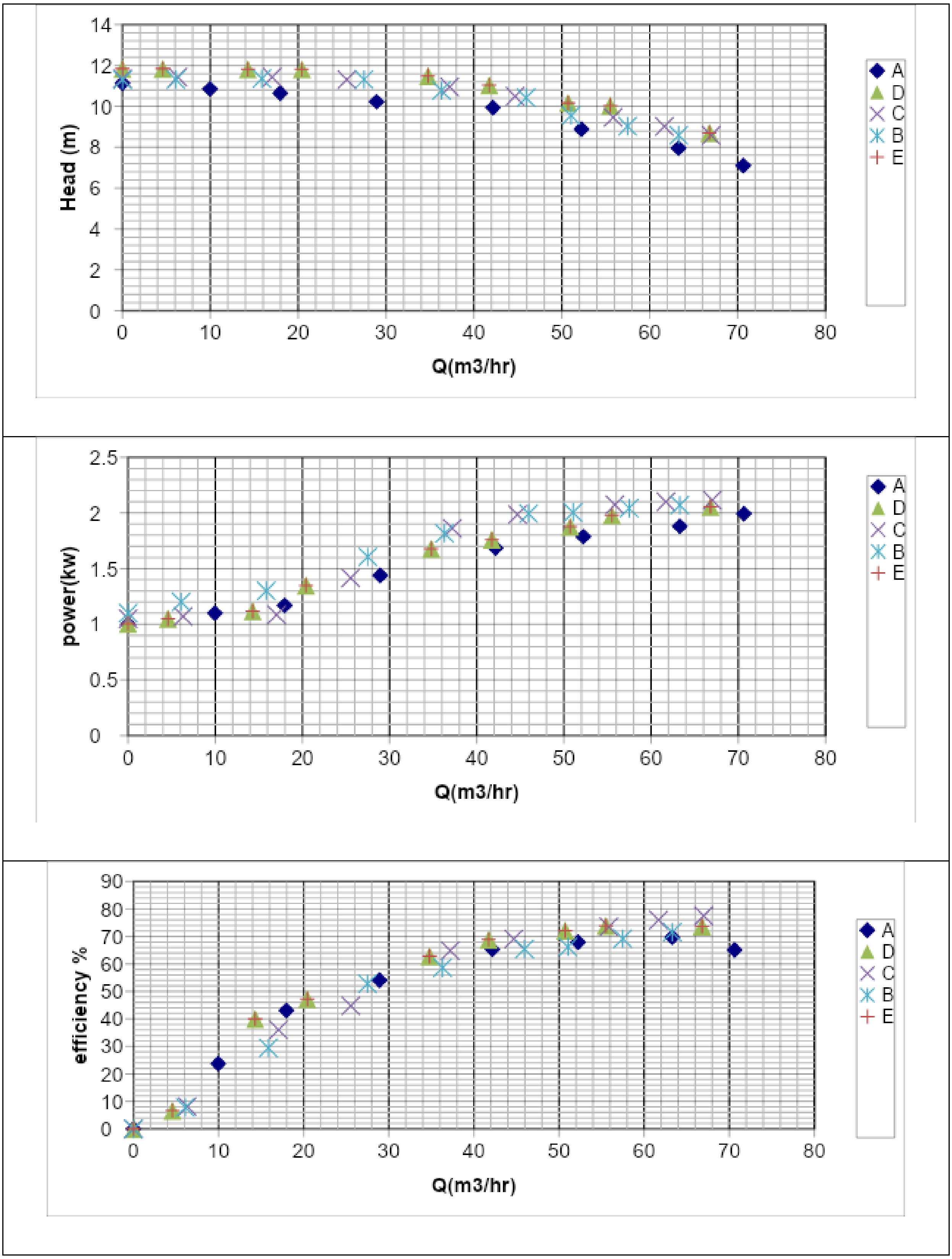


Fig 14 experimental results at 2000 RPM

IX. Conclusion

In the present work numerical and experimental study is attempted for centrifugal pump with original impeller and four modification impeller by varying the outlet vane angle of short blades installed between the main impeller blades. The numerical simulation using CFD predict the performance of the original impeller at different rotational speed with uncertainty different between the experimental performance not excised 10 %.the numerical results when adding short blades with the same main impeller outlet vane angle 21 º shows increasing in total head by 15% at nominal flow and 2900 RPM. The best improving in the total head is 17.9% at the smallest outlet vane angle 18º. The numerical results prove that the slip factor and total head increase when the value of outlet vane angle of short blades is decreased. The effect of short blades is clearly shown in the velocity magnitude leave the impeller which improves the value and direction of the fluid vector at this section. The experimental results of the laboratory tested pump shows the best improve for the total head by 3.2% for case with case the smallest outlet vane angle 18º at the nominal flow and 2900 RPM.

References

[1]. M.Gölcü, N. Usta, Y. Pancar, "Effects of Splitter Blades on Deep well Pump Performance", Jornal of Energy Resources Technology, Vol. 129/169, 2007, PP 169-176

[2]. M.F.Abd rabbo, M.S.Zahran, M.E.A.El Ghany,A.S.A.Zedan, and M.H.Shhata "Characteristic Of Centrifugal Pump With Shorted Blades".ENGNG.RES.JOUR.,VOL 83.PP.285-305, 2002

[3]. LIU Houlin, WANG Yong, YUAN Shouqi, TAN Minggao, a WANG Kai "Effects of Blade Number on Characteristics of Centrifugal Pumps".DOI: available online at www.cjmenet.com; www.cjmenet.com.cn 10.3901/CJME.2010

[4]. WEN-GUANG LI "Effect of Exit Blade Angle Viscosity and Roughness in Centrifugal Pumps Investigation by CFD Computation".Task Quarterly 15 No 1, 21–41 March 2011

[5]. Massinissa Djerroud, Guyh Dituba Ngoma, andWalid Ghie "Numerical Identification of Key Design Parameters Enhancing the Centrifugal Pump Performance: Impeller, Impeller-Volute,and Impeller-Diffuser".International Scholarly Research Network 10.5402/794341/2011.

[6]. Khalid. S. Rababa "The Effect of Blades Number and Shape on the Operating Characteristics of Groundwater Centrifugal Pumps". European Journal of Scientific Research ISSN 1450-216X Vol.52 No.2, pp.243-251,(2011).

[7]. Yang Sun-Sheng,1 Kong Fan-Yu,1 Fu Jian-Hui,2 and Xue Ling1 "Numerical Research on Effects of Splitter Blades to the Influence of Pump as Turbine " International Journal of Rotating Machinery 10.1155/123093/2012.

[8]. Wen-Guang LI "Blade Exit Angle Effects on Performance of a Standard Industrial Centrifugal Oil Pump". Department of Fluid Machinery, Lanzhou University of Technology 278 Langongping Road, 730050 Lanzhou, P R China.

[9]. M. F. Abd Rabbo, S. Sobhi, M. E. Abd EL-Ghany, A. Said Abd EL- Hafez and M. H. Shehata Effect of Shorted blade impeller on Radial Centrifugal Pump Characteristics. Personal contact.

[10]. said. A. F. Hawash, Abdullah H. I. Mohamed, and Mohamed H. Shehata, slip factor evaluation for centrifugal pump impeller with and without splitter blades, using CFD technique, International Journal of Innovative Research and Creative Technology, 2015 IJIRCT | ISSN: 2454- 5988.

[11]. Abdullah H. I. Aboelnil, Mohamed A. Abdellatif, Ibrahim Shahin, Mohamed A. Moawad, Mohamed F. Abd Rabbo, Performance and Slip Factor Prediction for Radial and Mixed Flow Pumps, International Journal of Scientific & Engineering Research Volume 10, Issue 7, July-2019 10 ISSN 2229-5518.

[12]. Moawad, M. A. and et. al.; “ Study of improving a centrifugal performance.“ Master Thesis, Shoubra Faculty of Engineering, Benha University, March, 2015, 88p.

[13]. E.C. Bacharoudis, A.E. Filios, M.D. Mentzos and D.P. Margaris, Parametric Study of a Centrifugal Pump Impeller by Varying the Outlet Blade Angle, *The Open Mechanical Engineering Journal,* 2008, *2*, 75-83.

[14]. A. Farid Ayad1, H. M. Abdalla, and A. Abou El-Azm, Effect of Semi-Open Impeller Side Clearance on the Centrifugal Pump Performance Using CFD, Aero space science and technology (2015,http://dx.doi.org/10.1016/j.ast.2015.09.033.

Abdullah.H.I.Aboelnil,etal. “Numerical and Experimental Study of a Centrifugal Pump with Varying the Outlet Short Blade Angle.” *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)*, 17(2), 2020, pp. 09-24.