CFD ANALYSIS OF PERFORMANCE CHARACTERISTICS OF VARIABLE SPEED DRIVE CENTRIFUGAL PUMP BY VARIATION IN NUMBER OF IMPELLER BLADES

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***Abstract*—*There is significant potential for energy efficiency improvement by using variable speed drive (VSD) centrifugal pump. In the field of centrifugal pump the impeller outlet diameter, the blade angle and the blade numbers are the most critical parameters which affect the performance. A CFD Analysis going to carry out on VSD pump at speed of 3000, 3500 and 4000 RPM by changing number of blades from 4 to 6 using ANSYS CFX software. The purpose of this analysis is to get performance characteristics of VSD pump by blade number variation. Also investigate how demand response can be improved using VSD system and the pump is to be used for wide range of demand response. Hopefully, this dissertation will be helpful to decide the number of blade to be used while designing impeller in VSD centrifugal pump.*

Keywords*— *Variable Speed Drive;CentrifugalPump;Demand Responce

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1. INTRODUCTION

A centrifugal pump is a dynamic device with the head generated by a rotating impeller. Thus, there is a relationship between impeller peripheral velocity and generated head. Peripheral velocity is directly related to shaft rotational speed, for a fixed impeller diameter. Varying the rotational speed therefore has a direct effect on the pump's performance. The equations relating centrifugal pump performance parameters of flow to speed, and head and power absorbed to speed, are known as the Affinity Laws as shown below:

$$\frac{Q_1}{Q_2} = \left(\frac{N_1}{N_2}\right); \quad \frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2; \quad \frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

Q_1 and Q_2 = discharge volume of inlet and outlet

N_1 and N_2 = pump speed at inlet and outlet

P_1 and P_2 = pump shaft input and output power.

Changing pump impeller diameter also effectively changes the duty point in a given system, and at low cost, but this can be used only for permanent adjustment to the pump curve and is not discussed further as a control method.

For systems where friction loss predominates, reducing pump speed moves the intersection point on the system curve along a line of constant efficiency. The operating point of the pump, relative to its best efficiency point, remains constant and the pump continues to operate in its ideal region.

The Affinity Laws are obeyed, which means that there is a substantial reduction in power absorbed accompanying the reduction in flow and head, making variable speed the ideal control method.

2. WORKING PRINCIPLE OF VARIABLE SPEED DRIVES

There are several types of VSDs, In applications that require flow or pressure control, particularly in systems with high friction loss, the most energy-efficient option for control is an electronic VSD, commonly referred to as a variable frequency drive (VFD). The most common form of VFD is the voltage- source, pulse-width modulated (PWM) frequency converter (often incorrectly referred to as an inverter). In its simplest form, the converter develops a voltage directly proportional to the frequency, which produces a constant magnetic flux in the motor. Figure 1 is sowing the schematic diagram VSD system. This electronic control can match the motor speed to the load requirement. This eliminates a number of costly and energy inefficient ancillaries, such as throttle valves or bypass systems.

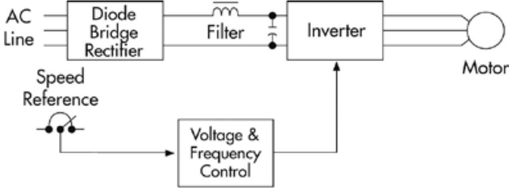

Fig. 1. Working principle of VSD system.

3. LITERATURE REVIEW

[1] R Menke at al 2017. Variable speed drive (VSD) pumps can be used to reduce the energy used to operate a Water Distribution System through the increased variation in the performance and saves energy up to 5%.[2] Abdellah kara omar at al 2016. Pump efficiency increases with decreasing rotational speed for flow rate values near 0.5 l/s. [3] Gideon Edgar Du Plessis 2013. VSDs were installed on pumps of the South Deep gold mine surface cooling systems for 1 month, and an electrical energy saving of 250MWh, or 29.9%, cost saving of US$11,996 and emission reduction of 227,893 kg were realized. [4] Rakibuzzaman at al 2015. In variable rotational speed the system curve would be cut the different pump head and the operation point of the pump can be operated below the constant drive pump. [5] Issa Chalghoum at al 2016 tested blades number of the impeller, staring time of t = 0.15 s, a

sharp difference between the pressure head curves for different blades number was obtained. These results have shown that the pressure evolution during the startup period is influenced by the valve opening percentages and the starting time. However, these parameters have no influence on the steady state pump characteristic curve. [6] Yanxia Fu at al 2014. The development and characteristics of cavitation in the test pump have been simulated at low, design, and high flow conditions, namely at 40%, 50%, 80%, 100%, and 120% of designed discharge.

At the low flow rate of 40% designed discharge the head-drop curves display a typical creeping shape just before breakdown conditions, quite different from the sudden head-drop occurring at high flow rates. It can be also indicated that the cavity volume grows when the inlet pressure is decreased and the shape of the cavities was nearly identical at low flow rates.

At high flow rates the structures of cavitation phenomena in the centrifugal pump are different from those at low flow rates. [7] Muhammad H Al-Khalifah and Gregory K. McMillan 2012. For pumping systems where the flow demand often drops, control valves will frequently be operating at lower throttle positions wasting more energy by a greater pressure drop across the valve.

Wasted energy due to pressure drop caused by discharge valve and flow recycling. Stiction, backlash, and other mechanical issues posed by control valve. Control valves have a dead band varying from 0.1% to 10% and a resolution varying from 0.05% to 5%.

Positioners have special tuning settings but rules and procedures are not well documented. Reducers, flanges, air supply lines, and isolation valves are required. Control valves are in the maintenance shop more than any other control device. Fugitive emission release from control valve packing.

VSD can increase process efficiency by reducing energy use and process variability. The savings are greatest for large flow systems, high turndowns, difficult process fluids, and extremely sensitive processes. [8] Dr. Mohammed ali mahmood hussein. 2013. The complex three dimensional internal flow field of the centrifugal pump is investigated, and following conclusion is attained.

The pressure increases gradually along the stream wise direction from minimum value at the suction side to the maximum value at leading edge.When the rotational speed increases, the pressure at the suction side decreases.

At the high rotational speed over the design point the pressure in the suction side became below the vaporization limits causes cavitation and wake region. [9] Vijaypratap R Singh at al 2014.Number of blades on the impeller number, Experiments shows that pump head increases with a greater number of blades. This is explained by the decrease in the liquid pressure drop in the flow passage with an augmented impeller blade number, keeping the same total volume flow rate the pump brake horsepower increases relatively with the augmented blade number. This is due to the increase in the pump shaft torque, as the pump blade number also increases. [10] S.Chakraborty at al 2011. With the increase in number of blades, the limitation of space between blade and flow stream gets increased. The area of low pressure region at the suction of the blade inlet grows continuously and the static pressure is gradually increasing with the increase in blade numbers. The uniformity of static pressure distribution at screw section becomes worse and worse, while at diffuser section, it becomes better and better. The head of centrifugal pump grows all the time with the increase of blade numbers and total pressure too, but the change in hydraulic efficiency with variation in blade number is little bit complex. [11] K.M. Pandey at al. 2012. In this paper the numerical analysis has been carried out for a number of impeller using different number of blades, but the impeller size, speed and blade angle being identical. with the increase of blade number the head is increasing at 2500 rpm rotational speed. With the increases of blade number, the head grows all the time, but the variable regulation of efficiency is quite complicated. [12] S R Shah at al 2013. CFD approach has been extensively used in centrifugal pumps as numerical simulation tool for performance prediction at design and off-design conditions, parametric study, cavitation analysis, analysis of interaction effects in different components, prediction of axial thrust, study of pump performance in turbine mode, diffuser pump analysis etc. [13] Yedidiah 2008. An improved design tool means that it provides a faster, more reliable, and less expensive means for achieving the targeted objective(s). In many cases, only CFD might be able to satisfy the above requirements, particularly in situations where a significant amount of trial and error work is required. At the present explosion of information, this requires close cooperation between the pump experts and the specialists in CFD. Properly applied, CFD has the potential of providing the practicing engineers with enormous assistance in their quest for better designs. **[14] M.Lorusso at al 2017.** In this work the numerical results were compared with experimental data. And CFX results are very accurate with respect to experimental data. The head increases asymptotically with the increase of the number of cells. This is due to the fact that, with greater number of elements, the accuracy of the simulation improves.

But to save computational time and to have a fairly accurate estimation of NPSH we can use lower number of elements.

4. RESEARCH METHODOLOGY

ANSYS software is used for the simulation of centrifugal pump at the rotating speed of 3000, 3500 and 4000 rpm by changing number of blades as 4, 5 and 6 at each speed. We obtained output results in the form of pressure developed at outlet and the overall efficiency of the pump. Obtained results are compared and came to the conclusion of selecting the blade which is most suitable for the pump at all speeds mentioned.

Boundary conditions

The pump has number of components like casing, blades, passage and wall interface in the model. P-total inlet and mass flow outlet boundary templates are considered in which the inlet is taken as a 1 atmospheric pressure and the mass flow outlet is taken as 6.94 kg/sec. Change in the speed gives better results in the form of pressure developed in the pump.

Inlet: It is at the eye of the pump as shown in fig. 2.

Wall: It is applied to Inlet passage faces, rotating faces of impeller and fixed faces of volute casing.
Outlet: It is applied on outlet face at end of the volute.
Fluid zone: It is applied at impeller passages, volute casing, inlet passage and outlet.
Solid zone: It is applied to impeller solid part.

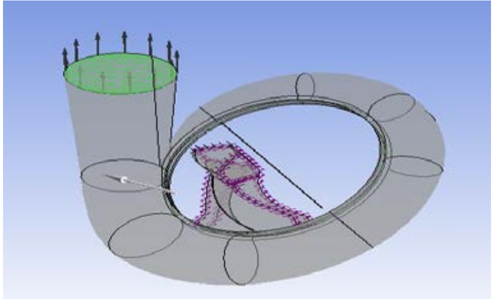

Fig. 2. Boundary condition of pump

Solution technique

The multiple reference frames (MRF) model is used to simultaneously simulate flow through rotating impeller and stationary volute casing. The impeller is modeled in a rotating reference frame and suction pipe, inlet passage and casing are kept as stationary reference frame. The model is used to simulate water transportation through pump and it uses a single fluid approach.

Post processing

The numerically simulated flow field of centrifugal pump was analyzed to investigate the performance characteristics of centrifugal pump at 3000, 3500 and 4000 rpm with number of blades at 4, 5 and 6. In this, the pressure contours were developed and information about the maximum pressure region developed is analyzed.

5. RESULTS AND DISCUSSION

On analysis in ANSYS simulation, we have plotted graph of each number of blades at all speed and flow rates. The graph we got is plotted as shown from fig. 3 to 5. And finally calculated overall efficiency and finally came to a result of selecting number of blades and speed.

TABLE 1 : CHANGE IN FLOW RATE, POWER AND PRESSURE OBSERVED AT OPERATING AT DIFFERENT SPEED AT 4 BLADES AND 32 HEAD

s.no.	Speed (RPM)	Flow (Q)	Power (kW)	Max. pressure (Pa)
1	3000	25	3.8	417320
2	3000	40	5.5	429221
3	3000	50	6.7	465310
4	3500	50	6.8	539328
5	3500	60	8	538702
6	4000	90	11.3	656920

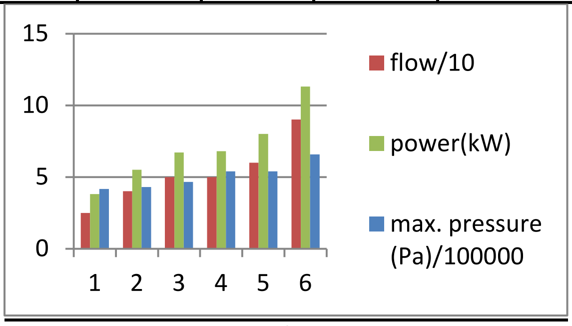

Fig. 3. Graph plotted from table 1 data

Table 2 : Change in flow rate, power and pressure observed at operating at different speed at 5 blades and 32 head

s.no.	Speed (RPM)	Flow (Q)	Power (kW)	Max. pressure (Pa)
1	3000	25	3.8	429272
2	3500	55	7.4	579293
3	3500	60	8	593431
4	4000	90	11.3	671549
5	4000	80	10.1	679689

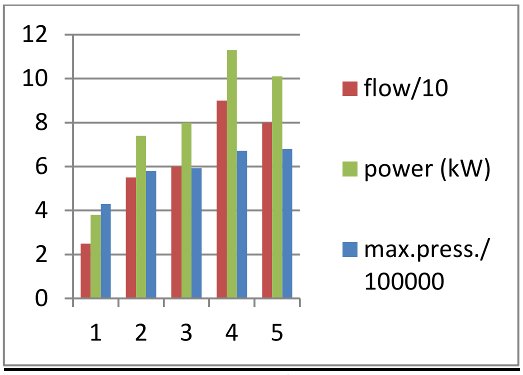

Fig. 4. Graph plotted from table 2

TABLE 3 : CHANGE IN FLOW RATE, POWER AND PRESSURE OBSERVED AT OPERATING AT DIFFERENT SPEED AT 6 BLADES AND 32 HEAD

S.no.	Speed (RPM)	flow (Q)	Power (kW)	max. pressure (Pa)
1	3000	25	3.8	430432
2	3000	40	5.5	451134
3	3500	60	8	588108
4	3500	70	9.1	665306
5	3500	80	10	579490
6	4000	80	10.1	725430
7	4000	90	11.3	698066

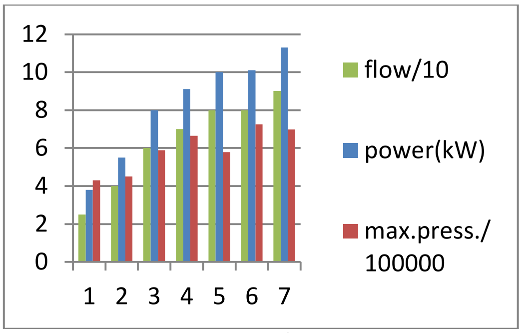

Fig. 5. Graph plotted from table 3

Using above data of table 1,2 and 3, we have finally came to a conclusion of finding overall efficiency and selecting how many blades to be used as shown in Table 4.

$$\text{Overall efficiency} = \frac{\rho \times g \times Q \times H}{1000\ P}$$

Here, $\rho = density\ of\ water,$ g = acceleration due to gravity, Q = rate of flow, H = head rise, P = power required to drive impeller

TABLE **4** : SHOWING OVERALL EFFICIENCY

Number of blades	**Speed (RPM)**	**Efficiency**
4	3000	0.6507463
4	3500	0.654
4	4000	0.6945133
5	3000	0.5736842
5	3500	0.654
5	4000	0.6945133
6	3000	0.6341818
6	3500	0.6976
6	4000	0.6945133

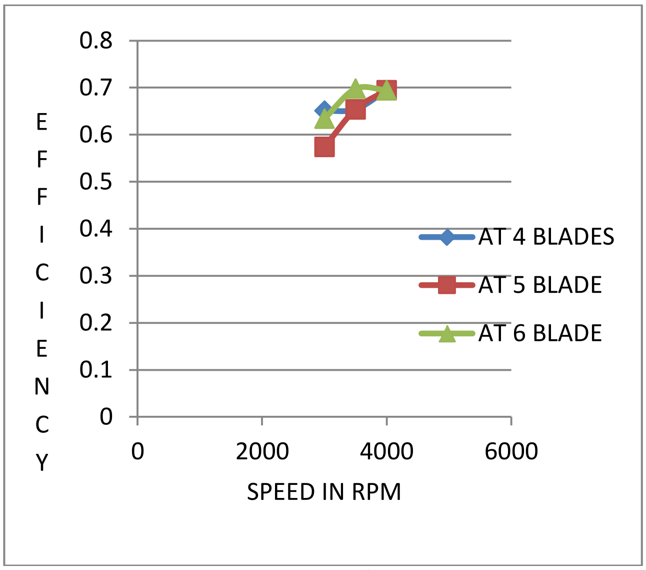

Fig. 6. Graph plotted from table 4

6. CONCLUSION

As from the fig 6, it is clearly admirable to use 6 blades and the operating conditions are needed to be as per the table number 11 to fulfill efficiency as well as demand response.

TABLE 5 : OPERATING CONDITION OF THE 6 BLADED CENTRIFUGAL PUMP

s.no	**Speed (RPM)**	**Flow (Q)**	**NPSH (m)**	**Power (kW)**	**min. pressure (Pa)**	**max. pressure (Pa)**	**efficiency**
1	3000	40	2.44	5.5	-48066.5	451134	0.6341818
2	3500	80	4.67	10	-57147	579490	0.6976
3	4000	90	6.04	11.3	-84086	698066	0.6945133

7. FUTURE WORK

We have validated this project from research paper and calculated and simulated all the data analytically and software based. There is no prototype made for the validation of this project. So my suggestion for future work is to make a prototype and validate the given parameters.

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