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A review on development in design of multistage centrifugal pump

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Abstract. Multi-stage pumps are the most popular pumps among various kinds of centrifugal pumps. A thorough review of different kinds of literature has led to the conclusion that there is a desperate need to increase the performance of the multi-stage centrifugal pump. Many investigators have put their efforts to increase the pump performance and also the work is being projected on various aspects of pump performance variables. To improve the multistage centrifugal pump performance by investigation, modification, and analysis many works of literature are available. For analysis, many researchers used the Navier-Stokes solver to create the three-dimensional unsteady turbulent flow numerical model with the standard k-ε turbulent equation. This paper mainly focuses on research related to the multi-stage centrifugal pump.

Keywords: hydraulic machine; modeling; multi-stage centrifugal pump; standard k-ε turbulent equation

1. Introduction

The Multi-stage centrifugal pump is widely used in agriculture, industries, domestic applications, and mining enterprises. The hydraulic machines which convert the mechanical

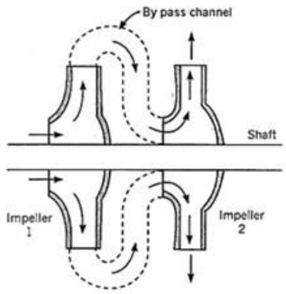

Fig. 1 Line diagram of a multi-stage centrifugal pump

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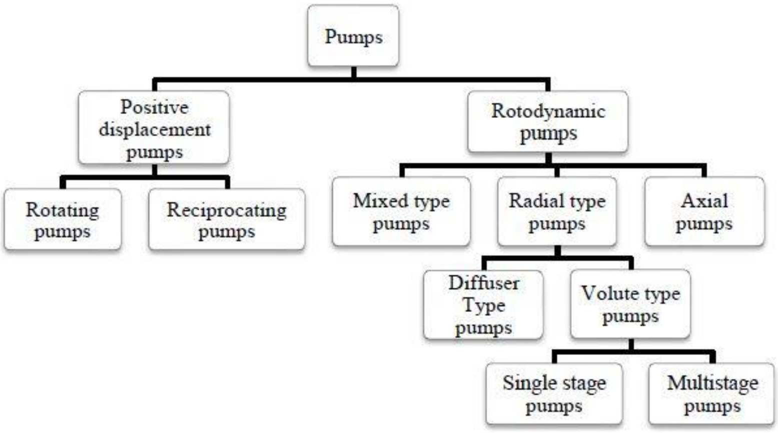


Fig. 2 Classification of pumps

(a) Open Impeller (b) Semi Impeller (c) Enclosed Impeller

Fig. 3 Types of impeller

energy into hydraulic energy are known as pumps and this hydraulic energy finds its way in the form of pressure energy, due to this raising of water from a lower level reservoir to a higher level reservoir. The simple centrifugal pump discharges the fluid up to a certain higher level reservoir since it does not develop sufficient pressure energy to raise the water from a lower level reservoir to a very higher level reservoir. In 1979, Gusher pumps manufacturer develops the multi-stage centrifugal pump to increase the head up to a very higher level by increasing the pressure in stages. It consists of two or more number of rotating impellers in a series attached to the same shaft. As there is an increase in the number of impellers, the head produce is higher than the simple centrifugal pump. The multi-stage pump used closed type impeller in which the impeller blades are enclosed between the shrouds. Fig. 1 shows the line diagram of the multi-stage centrifugal pump.

2. Classification of pumps

Based on the principle of operation pumps are classified into two types namely positive displacement pump and rotodynamic pump. A Positive displacement pump may operate by moving a fixed volume of fluid from the pump’s inlet pressure portion into the pump’s discharge

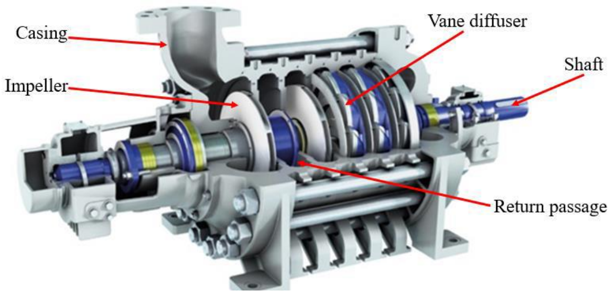
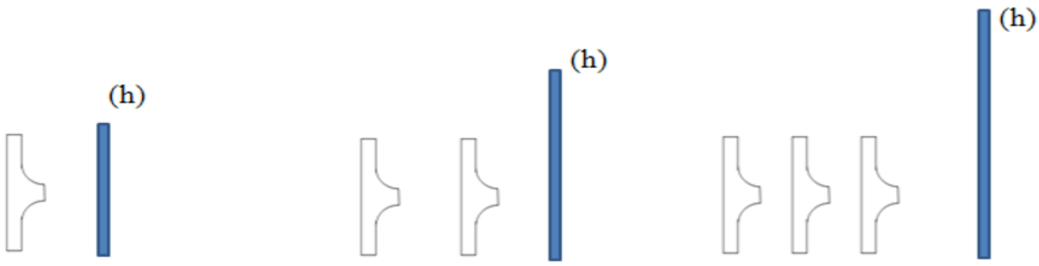


Fig. 4 The sectional view of multi-stage centrifugal pump



(a) single stage (b) two stage (c) three stage

Fig. 5 Principle of multi-stage centrifugal pump

region and a rotodynamic pump is a kinetic system in which a spinning impeller gives continuous energy to the pumped fluid. Fig. 2 shows the classification of the pumps.

3. Types of impeller

Depending on the type and the viscosity of the fluid to be impelled, the pump impeller is broadly categorized into three categories; open, semi-open and closed. Open impeller, where impeller only consists of blades and does not contain any shroud as shown in Fig. 3(a). Such pumps are mostly used in dredgers and the fluid containing solid particles. This type of impeller has to work under rough duty so generally it is made up of forged steel. The type of impeller where only one shroud and are used for viscous fluids like sewage water, paper pulp, sugar molasses, etc. is known as a semi-open impeller. This impeller has fewer blades and height is slightly more. Fig. 3(b) shows the semi-open impeller. Closed impeller the shroud is present on both the side of the impeller as shown in Fig. 3(c). Such an impeller is used to handle non-viscous fluid like water, hot oil, and chemicals, etc. Most of all, this sort of impeller is used in a multi-stage centrifugal pump (Lal J. (2005)).

4. Multi-stage centrifugal pump

A multi-stage centrifugal pump is a classification of a simple centrifugal pump. It consists basically of two or more revolving impellers arranged in a series attached on the same shaft and enclosed in the same casing. The main purpose of mounting the impellers in series is to attain a high head. Fig. 3 shows the view of a section of the multi-stage centrifugal pump.

Fluid first enters into the impeller of stage I then pass through a diffuser that is placed at the periphery of the impeller. After passing through a diffuser fluid take a sharp 90^0 turn to the back of the impeller and enters into the return passage where the return passage guide vanes guide the fluid to the next eye of the impeller. Fluid passes in a similar fashion through every stage of the pump.

As the machine enclosing a big number of impellers and has to produce high pressure the rotational speed required for such type of hydraulic machine is more than 1000 rpm. It is noted that the volumetric flow inside each impeller is equal. The use of this pump is found in reverse osmosis, spray, boiler feed, snowmaking, high-pressure draining, mine dewatering, etc.

4.1 Principle of multi-stage centrifugal pump

The multi-stage centrifugal pump is mainly used for high operating heads and the number of stages depends on the head needed. This is because there will be an increase in pressure in each stage of a pump, hence; therefore, total head developed is equal to the sum of the combined increase in pressure head of individual stage which is given by the following relation

$$\sum_{0}^{n} \Delta H = \frac{\sum_{0}^{n} \Delta P}{\rho g} \tag{1}$$

5. Literarture review

• The in-depth investigation by Gantar *et al*. (2002) on the hydraulic axial thrust problems of the multi-stage pumps, they focussed on the pump design concept having all impellers keyed to the shaft in a series. Laser Doppler Anemometry (LDA) is used to measure the fluid movement in the impeller side chamber. For the various leakage flow regimes, the fluid rotation phenomenon in the side chambers, and its hydraulic axial thrust effects has been examined. By introducing the Fluid Rotation Increasers (FRI) and Fluid Rotation Decreasers (FRD) into the impeller side chamber, the pressure distribution on the impeller front and back shroud had been manipulate which results in a reduction in the axial thrust. The sensitivity of the hydraulic axial thrust to the radial labyrinth wear is reduced much by the addition of the Multi Radial Bore wear rings (MRBwr).

• The numerical simulation was carried by Roclawski *et al*. (2006) on the radial multi-stage submersible centrifugal pump. The casing internal diameter (D_c) is much more than the impeller diameter (D_2) due to the presence of stator vanes i.e., ratio D_c/D_2 is more. So in order to minimize space, they built the pump with a small ratio of D_c/D_2 by keeping casing diameter (D_c) constant. Concerning space, the optimum ratio will be $D_c/D_2=1$. To realize this concept, before reaching the return stator the flow must exist the radial impeller in the axial direction. The Institute for Turbo Machinery and Fluid Mechanics at TU Kaiserslautern designed the alternative stators to reduce space and cost of the pump. They designed the pump in which the hub exit diameter of the impeller is reduced allowing the radial impeller to drain the fluid in axial direction before reaching the channel of return stator. Compared to traditional multi-stage pumps, the new stator pump efficiency is lower around 7% because of the losses that occurred in the pump stage. At the stator vanes leading edge (LE) and at the impeller suction side the vortices were developed due to extreme change of direction in the hub and also due to relatively thick stator vanes LE. The large separation between the flows is present in the return stator channels, which is also the reason of efficiency loss.

• Della Gatta *et al*. (2006) numerically found out the axial thrust balance in the horizontal multi-stage centrifugal pumps. In an industrial multi-stage machine to predict the axial thrust, there main target is to apply a three-dimensional CFD based tool. And to study the interaction between the rotating and stationary parts they choose the mixing plane technique. The results show that the axial thrust depends on the operating conditions of the pump as well as its mechanical wear conditions.

• Kawashima *et al*. (2008) experimentally investigated diffuser vanes effect on the multi-stage centrifugal pump performance by considering the interactions between the diffuser vane, the returning vane, and the next stage impeller. The results show that the vane profile of the diffuser used in the first stage influences not only the first stage but also the output of the second stage. It is also necessary that the diffuser vane matches well with the return vane to improve the pump's hydraulic performance. The performance of the pump may also be enhanced by increasing the channel sectional area from the outlet of diffuser to the inlet of the return passage as much as possible.

• The CFD analysis was carried out by Miyano *et al*. (2008) on return channel guide vanes They selected a two-stage pump with 7 impeller blades, 11 diffuser vanes, and 11 return channel guide vanes and k- ε model was used to solve the problem. They disclosed that if the return passage vanes trailing edge is set to the downstream annular channel outer wall radius, the fluid will stop swirling before reaching the next stage, which results that the next impeller is being able to draw more fluid and add more power to the fluid.

• Tverdokhleb *et al*. (2012) investigated the development of the multi-stage pump flow component with minimum radial diamensions. To maintain a high level of energy performance by reducing the radial dimensions the issue of a specific quantity of metal of multi-stage pump reduces. The first possibility is to reduce the diameter of guide vane, but there is a specified limit to reduce the diameter of the guide vane (D_{GV}), after which the efficiency of guide vane will sharply fall which is highly undesirable. The statistical studies show that at the intermediate stage hydraulic efficiency is in the range of 0.87 to 0.98. The second possibility is to reduce the diameter impeller outlet (D_2), while maintaining the required pressure head the hydraulic efficiency is in the range of 0.96 to 0.98.

• Lugovaya *et al*. (2012) revised the problem that is persistent in the designing of an intermediate stage guide-vane of centrifugal pumps. They use two types of guide vanes viz. guide vanes with continuous transferable channels (CTC) and with the intermittent transferable zone (ITZ). Guide vane with CTC has large mass and radial dimensions at the same stage parameters. At constant rotational speed, to minimize the cost of the multi-stage centrifugal pump there should be reduced in its weight and size characteristics and guide vane provide with ITZ.

• The impeller of the centrifugal pump is analyzed by Rajendran and Purushothaman (2012) with the help of ANSYS-CFX. They analyzed the flow pattern, pressure distribution inside the flow passage, blade loading and the pressure plots. The result shows that continuous pressure increase occurs from leading edge to trailing edge of the impeller. This is because the rotating pump impeller had developed a dynamic head. Because of the blade thickness at the leading edge low pressure and the high velocities are observed and at the trailing edge of the blade total pressure loss is observed in the vicinity of trailing wake.

• Roche-Carrier *et al*. (2013) investigated the first stage of a multi-stage centrifugal pump numerically. It consists of impeller, diffuser with return vanes and casing. They carried out CFD analysis by using k-ε turbulence model. The analysis is conducted with five parameters i.e. by varying impeller blade height, by varying impeller number of blades, by varying diffuser vanes, by varying return vanes, and by varying wall roughness. The results show that when the impeller blade height was 29 mm the head obtained was found to be highest but the height 23 mm gives the best efficiency point. By varying the impeller blades numbers, impeller with 7 number of blades gives the maximum efficiency and also it is observed that as impeller blade numbers increase head and performance of the pump improved. By varying the number of diffuser vanes, the highest head and efficiency were found for 7 and 8 diffuser vanes respectively. By varying return vanes, highest head and efficiency were given by 11 return vanes. It is found that pump with 7 impeller blades, 8 diffuser vanes & 11 return vanes has the best efficiency. The variation in efficiency is not affected by wall roughness smaller than or equal to 0.002 mm, but it starts decreasing for higher values of wall roughness.

• The multi-stage centrifugal pump model is analyzed by Naveena and Suresh (2013) using the finite element method (FEM) technology for catching natural frequencies from the centrifugal rotor. No resonance was found for all 3600 rpm to 5130 rpm operating speeds. Whereas the maximum amplitude of the deflection of the rotor for the unbalance loading is determined. They found that the deflection of the rotor for unbalance loading is 7.12% which is less than 35% of the diametric clearance as defined by API 610 standards.

• Wang *et al*. (2013) predicted the 3-D unsteady turbulent flow in a multi-stage centrifugal pump. The 3D unsteady state turbulent flow numerical model is modeled which is based on Navier-Stoke solver and standard k-ε turbulent equations. The results indicate that the flow in the impeller is mostly uniform, with some eddies, backflow, split flow, and the jet-wake phenomena, which occurs only along with individual blades. It was also observed that the secondary flow is present at the end of the blade and the outlet of the guide vane. The total pressure distribution remains asymmetric due to different blade, several guide vanes, and impeller.

• Quality features of a centrifugal pump with forward, radial and backward bladed types of the impeller are reported by Marathe *et al*. (2013). For the analysis purpose, the k-ε turbulence model is used. From the results, it was found that for low liquid head the pump works efficiently when worked with an impeller bladed backward and also the problem of cavitation is less.

• Rakibuzzaman *et al*. (2015) investigated the multi-stage centrifugal pump performance characteristics using the inverter control variable speed drive system in order to get energy saving rates instead of the constant speed drive system. In the experimental setup variable voltage variable frequency vector controlled inverter drive was installed. The result shows that at the same speed ratio when a similar type of pumps would be running the energy saving could be obtained. For the validation of results, the three pumps system head performance was also carried out in parallel to compare with one pump system head. The operating point drops down the system curve for variable speed drive pump with the result that the flow and head were reduced accordingly at controlled speed. The operating point moves forward to the head curve in case of constant drive pump as a consequence the flow is reduced and the head increased. The point of interaction of the operating point is the only place where the flow rates of the pump and the system are equal and the head of the pump and the system head are also equal simultaneously.

• Zang *et al*. (2015) investigated the inside flow of a multi-stage centrifugal pump based on the 3D unsteady RANS equations and the standard k-ε turbulence model. The result indicates the lowest pressure area where cavitation is easy to occur, found in the inlet of the back of the first impeller blade. At the return guide vane, an area of low pressure can be observed. Also, velocity and static distributing pressure show asymmetry and the secondary phenomenon were formed near the impeller outlet.

• Chhanya (2015) reviewed the impeller of a mixed flow type submersible pump to improve pump performance. They contrasted the current impeller configuration with the best efficiency point of the submersible pump impeller mixed-flow model. The results show that the performance of the pump can be significantly improved, with the change in blades inlet and outlet angle and by reducing the number of blades. Almost 12% head has been improved with a constant discharge rate of 0.0078 m^3/sec by improving the design of the impeller at the best efficiency point. Due to this, hydraulic efficiency is improved by 20% in the case of experimental work whereas the efficiency was improved by 25% in the case of numerical simulation.

• Tan *et al*. (2016) studied the effect of vane angle, vane thickness, and the number of vanes of radial diffuser for a single-stage centrifugal pump having Q=32m^3/h, speed= 1450 rpm and H=7.5 m. For the design of the vane the single arc method is used, the analysis was carried out for different angles at the outlet (3°, 6°, 10°, 15°, 25° & 40°), the results show the maximum efficiency and head were achieved by the 6°, while 40° resulted in a minimum value among all. Among 2 mm, 5 mm, 8 mm & 10 mm vane thickness, the vane with a thickness of 2 mm gives better results. The performance was unsatisfactory while analyzing the number of vanes with the vanes less than 8 or greater than 8 and the losses were lowest when there were 8 vanes.

• The pump output characteristics for different radial diffuser shapes are numerically analyzed by Kim *et al*. (2017). The specification variables selected are the number of vanes, return channel vane trailing angle (α_6), diffuser diameter ratio (D_4/D_3), and pressure recovery factor (C_p) for the radial diffuser. For the analysis, the number of vanes was selected as 7 for the impeller and 10 for the radial diffuser. It is found that the head loss is lower by 0.2% only when the return channel outlet angle (α_6) is 90°, therefore, the efficiency of the pump is higher when α_6 is 90°. There will be higher loss in the head when the α_6 is 60° due to the higher pre-swirl at the inlet of next stage impeller so the efficiency of the pump decreases. Head losses were compared for different ratio D_4/D_3, it can be seen that the C_p increases as a ratio D_4/D_3 increases. As C_p increases the head loss factor decreases which results in high head and efficiency.

• Bai *et al*. (2017) worked on the matching of impeller blades and diffuser vanes. They did the analysis for various combination of impeller with blades 5, 6 & 7 and diffuser with vanes 8, 9 & 10. The problem was solved with the help of 3D unsteady RANS with SST k- ω turbulence model. For both numerical and experimental studies, they used pump of 6000 rpm, Q=200 m^3/h, N_s=71.6 and design head=300 m. The lowest value of head was found for the combinations of 5+8, 6+8, 7+9. By keeping the impeller blades 7 constant, for 8 diffuser vanes the efficiency was found maximum and it is minimum for 10 diffuser vanes. They showed that more will be the passage area if the diffuser vanes are less in numbers, which results, less blockage and less fluctuation. When the number of impeller blades (5, 6 & 7) changes and the other parameters remains constant the head is increased by 2.47%, 0.71% and 1.39% and efficiency increased by 2.77%, 3.02% and 3.77%, respectively. They suggested that for steady and unsteady results the pump with 7 impeller blades and 8 diffuser vanes is the best choice.

• Wang *et al*. (2019) had examined the effect of the turbulence model on the five-stage centrifugal pump sound field. The standard *k*-ε and STT models were used to simulate the flow and sound fields of a five-stage centrifugal pump with a vane-diffuser. The results indicate that under three flow rates, the peak value of pressure pulsation occurs at 444 Hz. There is a rise of a pressure pulsation amplitude with an increase in the flow rate. For the internal sound field, the sound power (L_p) at the suction stayed at 108 dB and at the discharge it stayed at 114 dB, whereas 79-87 dB of L_p was maintained for the analysis of the external sound field.

6. Conclusions

In this review, various parts of literature that suggest the possible improvement in the multi-stage centrifugal pump performance are studied. More vividly the flow in the impeller is assumed to be uniform.

- The issue of a specific quantity of metal reduces by reducing the radial dimension of the pump.
- The axial thrust on the machine, which is now considered as a pump depends on its working condition and its mechanical wearing condition.
- The sensitivity of the hydraulic axial thrust to the radial labyrinth wear is reduced much by the addition of the Multi Radial Bore wear rings (MRBwr).
- At constant rotational speed, the cost of the pump can be minimized considerably by reducing its weight and size characteristics.
- As the height of impeller blades, diffuser vanes and the number of blades increases the pump stage head and brake horsepower increases.
- The performance of the pump may also be enhanced by increasing the channel sectional area from the outlet of diffuser to the inlet of the return channel as much as possible.
- At the stator vanes leading edge (LE) and at the impeller suction side the vortices were developed due to extreme change of direction in the hub and also due to relatively thick stator vanes LE.
- The return passage vanes trailing edge is set to the downstream annular channel outer wall radius, the fluid will stop swirling before reaching the next stage, which results that the next impeller is being able to draw more fluid and add more power to the fluid.
- The large separation between the flows is present in the return stator channels, which is also the reason of efficiency loss.
- The variation in efficiency is not affected by wall roughness smaller than or equal to 0.002 mm, but it starts decreasing for higher values of wall roughness.
- The lowest pressure area where cavitation is easy to occur found in the inlet of the back of the first impeller blade and at the return guide vane an area of low pressure has been observed.
- The diffuser vane with a thickness of 2 mm gives better results and the performance was unsatisfactory while analyzing the number of vanes with the vanes less than 8 or greater than 8 and the losses were lowest when there were 8 vanes.
- The pressure recovery factor (Cp) increases as a ratio D_4/D_3 increases and as Cp increases the head loss factor decreases which results in high head and efficiency.
- For steady and unsteady results the pump with 7 impeller blades and 8 diffuser vanes is the best choice.
- The peak value of pressure pulsation occurs at 444 Hz and there is a rise of a pressure pulsation amplitude with an increase in the flow rate.

By the thorough review of the literatures it is found that the performance of the pump is mostly affected by cavitation and there are various ways to increase the efficiency of the multi-stage centrifugal pump; but still simpler modelling approach will help the scientific community in understanding the dynamics of the pump.

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Appendix

Nomenclature

n	Total number of stages
ΔH	Increase in Head in each stage (m)
ΔP	Increase in pressure in each stage (N/m^2)
ρ	Density of liquid (kg/m^3)
g	Acceleration due to gravity (9.81 m/s^2)