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Research Article

Critical Hydraulic Eccentricity Estimation in Vertical Turbine Pump Impeller to Control Vibration

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In many applications, pumps are tested against standard specifications to define the maximum allowable vibration amplitude limits of a pump. It is essential to identify the causes of vibration and methods to attenuate the same to ensure the safe and satisfactory operation of a pump. Causes of vibration can be classified mainly into mechanical and hydraulic nature. Respective unbalance masses are the two major factors which cause dynamic effects and excitation forces leading to undesirable vibrations. In this paper, the procedure of vibration magnitude measurement of a vertical turbine pump at site and the process of dynamic balancing to measure mechanical unbalance of an impeller are explained. After that, the impact of hydraulic eccentricity on the vibration displacement of a vertical turbine pump has been explained using numerical simulation procedure based on "One-way Fluid Structure Interaction (FSI)." The experimental results from a pump at site are used to compare the numerical results. After the solver validation, the one-way FSI approach is used to find the critical hydraulic eccentricity magnitude of a vertical turbine pump impeller to limit the vibration magnitudes on motor component to less than $100\,\mu$m. From the numerical simulations, it is deduced that the critical hydraulic eccentricity should be limited to $400\,\mu$m in X and Y direction. The process can be used as a guideline procedure for limiting the hydraulic unbalance in vertical turbine pumps by limiting the hydraulic eccentricity.

1. Introduction

Geometrical deviation in any impeller is due to the manufacturing process. It is very difficult to produce an ideal component with design dimensions. Hence, the tolerances are defined for the manufacturing process to ensure a safe and satisfactory operation. The existing process in any industry is to have a final component within the specified tolerances. A pump impeller with geometrical deviation due to the manufacturing process causes two types of unbalance: (1) mechanical unbalance and (2) hydraulic unbalance. The mechanical unbalance can be corrected in machine shop by removing or addition of material to align the mass center of the component with its center. However, no traditional methods are available to eliminate the hydraulic balance in a machine shop. The prevalent reasons for excessive pump vibrations are due to rotor dynamics, fluid dynamics, and structural type. The causes for the vibrations can be understood by studying the data of the vibration readings over a range of flow, speed, and other variables such as pressure and temperature. The rotor dynamics causes can be studied using spectrum analysis and Bode plots. The structural resonance can be found using modal analysis. The fluid dynamic vibrations can be studied using spectrum analysis under different operating conditions. However, the prediction of fluid dynamic vibrations is very difficult to find during testing [1]. Dynamic balancing of a rotating system can be used to limit the mechanical unbalance, which is a direct method. However, no direct method is available for hydraulic unbalance. The reason for vibration in a pump unit ranges over a broad range of causes, which includes the pump type (radial, mixed flow, or axial), operating points at the site, system resistance, type of fluid, design of impeller, and flow intake conditions. The other important phenomenon which

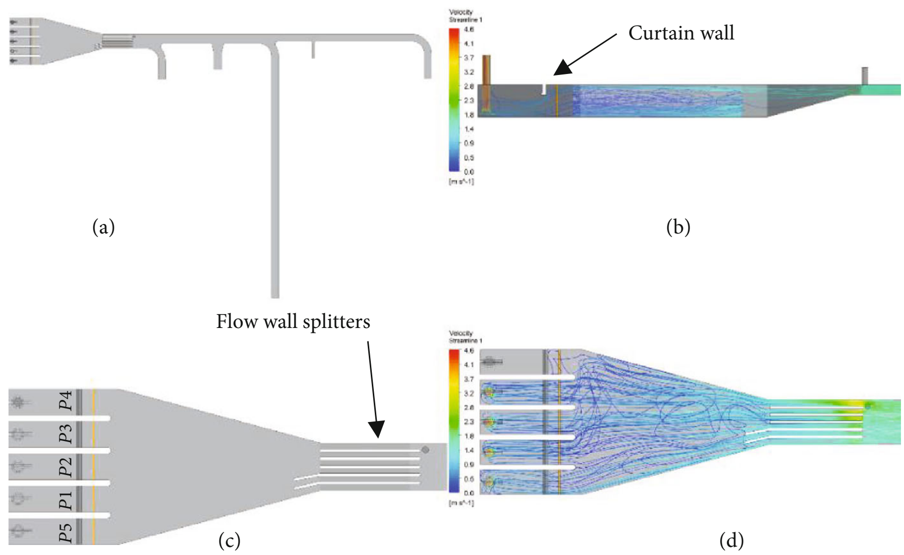
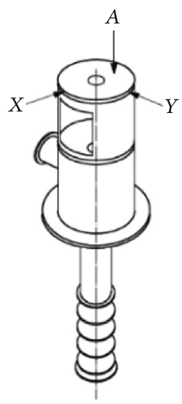


Figure 1: Pump intake—geometry and streamlines.

can cause vibrations is suction cavitation. All these factors are to be considered for the possible causes of the severe vibrations which may further lead to shaft failure and hence the operational loss to the customer [2]. Apart from the individual components, the coupled impact also affects the vibrations. A theoretical study has been conducted for an impeller under two-dimensional conditions. It concluded that the flow nonuniformity created by a rotating impeller affects the volute casing performance [3]. At sites, FFT also used to measure the vibration displacements and predict the possible cause of failure. Vertically suspended pumps have been analyzed to find the root cause of the vibrations [4]. The upstream flow conditions of a pump suction or bell mouth are defined by the pump intake design. The possible adverse flow phenomena near pump suction due to improper intake design are high swirling flow, surface vortices of different types, and air entrainment [5]. The improper intake design enhances the magnitude of vibrations in the pump due to the resulting nonuniform flow condition at the suction side. Additionally, if there is any presence of manufacturing deviation in a rotating geometry, it causes the increase in the magnitude of the unbalance force. The unbalance force magnitude raises with the raise in deviation from design geometry due to the manufacturing process [6]. The permissible residual unbalance in terms of eccentricity for a rotating impeller has been given in the form of a graph in ANSI HI standard [7]. Finding the cause of vibration for any pump system at the site is not a single step method. It is a stepwise method to trouble shoot vibration problem at site. The study of pump system is recommended to find the cause and provide the solution. From the manufacturing point of view, any deviation in the impeller design results in enhancing the subsynchronous vibrations which further lead to shaft failure in vertical turbine type pumps [8]. The operation of a vertical turbine pump depends on mainly on three factors which are suction conditions which are the pump intake design, discharge piping design, and the system resistance characteristics. Accordingly, any vertical turbine pump pumping system can be divided into three parts: (1) pump intake, (2) pumping system, and (3) discharge system [9].

Different parts of a system-related flow characteristics and its impact are always coupled in a vertical turbine pump. The flow-induced forces in any part cause vibration of the structure, and the structural vibrations are further impacting the flow characteristics in return. Few of the fluid-structure interactions in a vertical pump are vortex-induced vibrations, flow-induced vibrations in annular sections, and flow instability due to leakage flow. These phenomena are further verified with tests at different conditions [10]. The impeller unbalance has been divided into two types: (1) mechanical or structural unbalance and (2) hydraulic unbalance. Mechanical unbalance is purely related to the manufacturing deviation. It is defined as the deviation between the mass centerline of a rotating impeller and the shaft axis. Hydraulic unbalance is defined as the flow unevenness between impeller vanes, which is due to geometrical deviations in vane pitch. The nonuniform vane pitch in an impeller which occurs during the manufacturing stage results in the hydraulic unbalance. This unbalance force rotates at rotor speed. Apart from the speed of the impeller, the flow rate also increases the unbalance force [11].

Numerical tools and simulations are playing a major role in the present digital world to predict the design product efficiency prior to its manufacture. It also helps in studying the product under typical operating conditions and viewing the output parameters at different points, which is a cumbersome effort during testing. In centrifugal pump industry, numerical simulations are used in many applications, which include performance prediction, cavitation and erosion level prediction, natural frequency of a system, and rotor dynamic analysis. It is also possible to check the variation of different parameters and its impact on the performance using numerical tools. Numerical simulations are also having vital importance in pump intake design, impeller design, and system design. The primary output parameters from the simulations are performance prediction, flow phenomenon prediction,

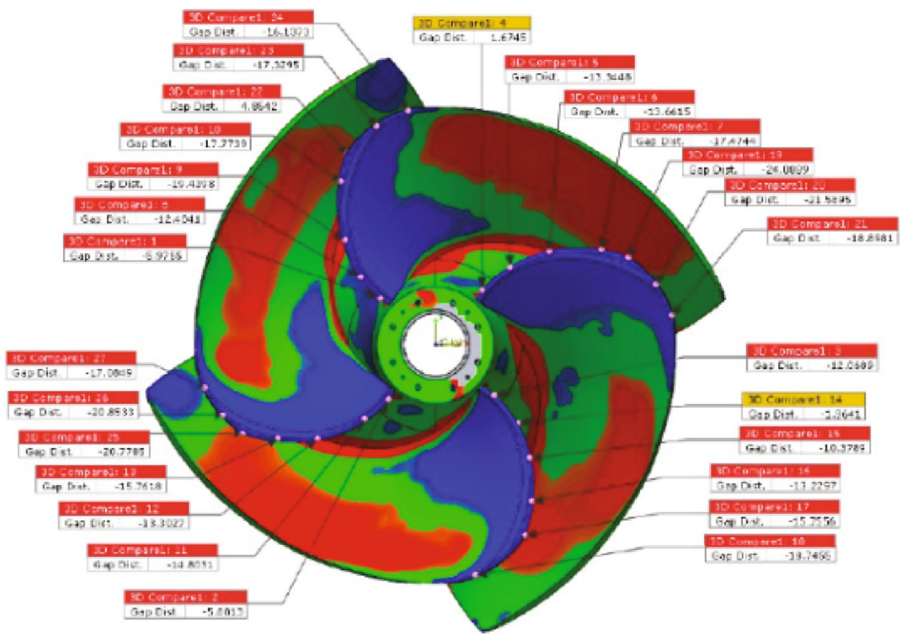


(a)

(b)

(c)

Figure 2: Continued.



(d)

Figure 2: (a) Pump vibration measurement location. (b) Pump at site. (c) Impeller. (d) Scanned image of impeller.

and cavitation prediction. CFD tools are widely used by industries to check the design and do the required modifications [12]. The recent developments in CFD tools include the coupled flow simulation which describe the real flow phenomenon. The coupling can be one way or two ways based on the nature of the problem. Both methods have various advantages in terms of accuracy and resources and applicable for certain applications [13].

The annular regions in the seals are with very high pressure magnitudes, and the coupled flow analysis gives the appropriate flow characteristics. The L/D ratio and impact in annular seals are well predicted with FSI technique, and its impact on critical speed is found to be aligned with the test data [14]. The flow-induced forces are also possible due to upstream flow conditions and which can be limited by a proper intake design. The suction recirculation is the main phenomenon result from improper intake design. In such cases, the provisions like splitter walls at the back wall and bottom wall and the change in the design of the impeller in terms of the number of blades reduce the suction recirculation [15]. Moreover, the upstream flow impact will be enhanced after the flow passes over the impeller-to-discharge system. The relative impact of the discharge system can be well predicted by fluid-structure interaction. The numerical simulation results are found to be aligned with the test data [16]. The Modal Analysis of Vertical Turbine Pump has been conducted for predicting the natural frequencies of the pump and to understand its mode shape. The structure can be modified or stiffened at appropriate locations based on the natural frequency margin over the operating speed of the pump [17]. The suction recirculation further results in the cavitation of the impeller due to creation of low-pressure zone. This results in pitting over the impeller surface

Table 1: Test data of vibration displacements.

Vibration readings on vertical turbine pumps, displacements (μm)						
Pump location	Motor NDE		Motor DE		Pump DE	
number	X	Y	X	Y	X	Y
1	62	51	49	34	28	45
2	148	159	404	376	281	254
3	99	119	59	61	55	41
4	58	41	37	29	22	41
5	77	72	64	49	35	34

and over a period of time changes the hydraulic profile of the vane. This continuous phenomenon causes excessive vibrations in the rotating unit due to flow-induced unbalance forces near the impeller with uneven mass distribution [18]. The trouble shooting of vibration problems in vertical pumps and the solution for the same is possible with the available numerical tools. The impact of various structural modifications on structural resonance is possible to find with optimum resources. The stiffness of the prime mover base has been increased numerically and found that the resonance has been shifted from 1X operating speed [19]. Moreover, the flow coupled simulations are used to find the cause of vibration in a vertical pump. Hydraulic unbalance is a phenomenon which is possible to check only either by theoretical method or numerical tools. The one-way fluid-structure simulation of a vertical turbine pump with hydraulic unbalance and its impact on vibration displacements have been predicted and shown good agreement with the experimental test data [20].

The mechanical eccentricity is considered during dynamic balancing of rotating system. It is important to

Table 2: Details of mesh statistics of four test cases.

Mesh	Bell mouth	Impeller	Bowl	Column pipe	Total nodes (million)	Head (m)	Absolute deviation with test data (%)
Mesh 1	0.017	0.23	0.31	0.12	0.68	22.6	—
Mesh 2	0.02	0.265	0.365	0.17	0.82	23.1	2.2
Mesh 3	0.0287	0.32	0.415	0.206	0.97	23.5	1.7
Mesh 4	0.033	0.375	0.47	0.235	1.12	23.6	0.5

consider hydraulic unbalance which cannot be balanced by dynamic balancing method on balancing machine. Hence, the purpose of the present study is to understand the impact of hydraulic unbalance on the vibration displacements of specific rotor and stator components of a vertical turbine pump. In addition, numerical studies are used to provide the preliminary idea in terms of eccentricity of hydraulic masses to limit the vibrations. First, the analytical method (theoretical and graphical) is reported to estimate the eccentricity of hydraulic rotating masses rotating in a single plane. Second, to predict the impact of eccentricity of the hydraulic portion on vibrations, numerical simulations were conducted. ISO 1940-1:2003 was used as an experimental method to estimate the vibration displacement limits of certain stator and rotor components. One-way fluid-structure interaction method is used to estimate the impact of the eccentricity on the vibration displacements of specific parts. The obtained numerical results are compared with the test results and found that the results are in good agreement. Finally, the eccentricity of hydraulic mass is used as a variable in numerical simulations to estimate the impact of eccentricity of hydraulic mass which leads to excessive vibrations in the pump components.

2. Materials and Methods

2.1. Pump Intake Geometry. At site, there are totally five vertical turbine pumps for which the pump intake has been designed. Each pump is to deliver flow of 30000 m^3/h with 23.5 m head at the speed of 373 rpm. Figure 1 shows the geometry (a, b) and flow streamlines (c, d) of the modified pump intake geometry. The sump geometry is having multiple inlets to flow in the perpendicular direction to the fore bay wall. Pumps located in the pump chamber are notated as P_1, P_2, P_3, P_4, and P_5 for measurement purposes. P stands for pump, and suffix number stands for pump number. Curtain wall at the appropriate location in the original geometry, the guide walls, and flow wall splitters in the pump chambers are finalized using CFD analysis for flow improvement in the sump, and the same have been incorporated at site.

2.2. Pump Vibration Problem. Excessive vibration was observed at the site for two pumps among 5 pumps. To check the cause of the vibrations in two particular pumps and to reduce the magnitude of the vibration in respective pumps, a step-by-step trouble shooting procedure has been applied. Below subsections describe the vibration measurement procedure along with the different phenomenon related to pump vibration.

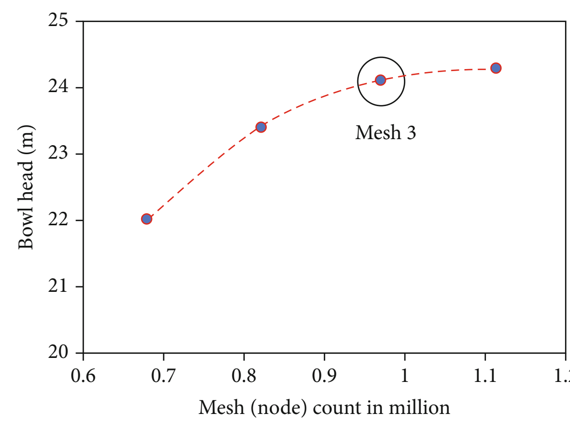


Figure 3: Mesh independent study (CFD) of the pump.

2.3. Pump Vibration Measurements at Site. The vibrations are measured at motor top location as indicated in Figure 2(a) in X, Y, and A directions. X and Y are vibrations in the lateral direction, and A is in the axial direction. Vibrations are measured using NPP-KOHTECT make C911 vibration analyzer with magnetic base type accelerometer. Accelerometers are mounted at respective locations to measure the vibration in terms of displacement units being slow speed machine (less than 600 rpm speed). Vibrations are measured at Motor Non-Drive End (MNDE) bearing, Motor Drive End (MDE) bearing, and Pump Drive End (PDE) bearing location. The vibration response is obtained by capturing the vibration displacement data while pump is in operation.

2.4. Pump Overhauling. Overhauling is a procedure to disassemble the pump components and inspect the individual components to replace and reassemble the pump. For the pumps with high vibrations, the overhauling procedure has been carried out to check the impeller, bowl unit, shaft, shaft sleeves, bell mouth, suction side bearing, and shaft intermediate bearings.

2.5. Dynamic Balancing of Impeller. The residual mass unbalance in any rotating system creates a force which further transmits to the bearings and foundation unit when it rotates at rated speed. To minimize the forces, the balancing is to be carried out to limit unbalance mass by referring dynamic balancing procedure. For the present case, balancing of the rotating units comprising of parts like impeller, shaft, sleeves, and keys is carried as per standard ISO 1940 grade 6.3.

2.6. Repeated Vibration Measurements. Vibration measurement is conducted for 5 pumps at site, and Table 1 shows

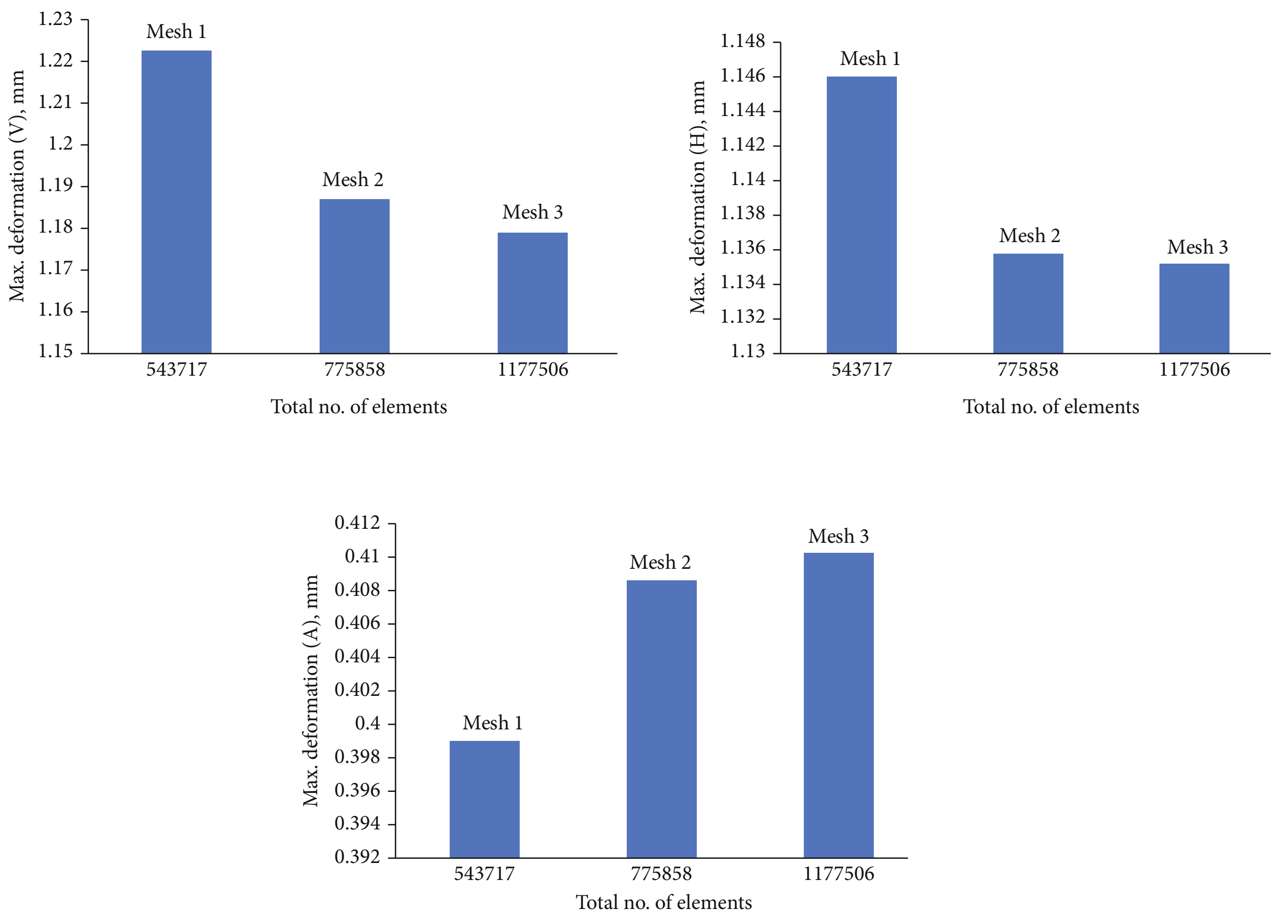


Figure 4: Mesh independence study plots of FEA model.

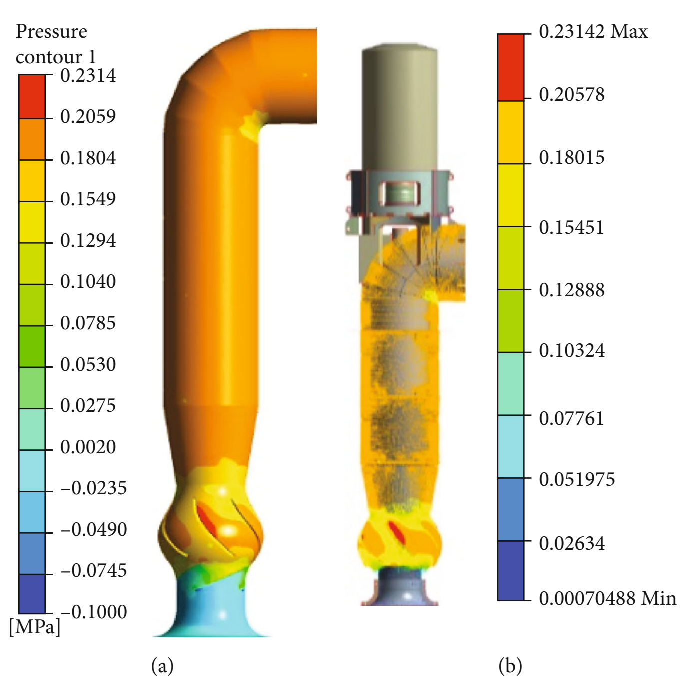


Figure 5: Pressure contours from CFD model to structural model: (a) CFD model; (b) structural model after pressure mapping.

Table 3: Vibration displacements when impeller is with hydraulic unbalance.

Sr. no.	Hydraulic eccentricity 688 (μm)	Bell mouth	Shaft	Impeller	Suction bearing	Discharge bend	Motor
Maximum vibration displacement X (μm)							
1	Numerical results	296	239	222	173	57	137
2	Test results	—	—	—	—	—	148
Maximum vibration displacement Y (μm)							
1	Numerical results	364	298	279	222	64	141
2	Test results	—	—	—	—	—	159

Table 4: Vibration displacement when the impeller is on hydraulic unbalance.

Sr. no	Hydraulic eccentricity (μm)	Bell mouth	Shaft	Impeller	Suction bearing	Discharge bend	Motor
Maximum vibration displacement X (μm)							
1	0 (design)	166	78	74	60	20	44
2	200	204	125	117	93	30	71
3	300	223	149	139	109	36	84
4	400	242	172	160	126	41	98
5	500	261	196	182	142	47	112
6	688	296	239	222	173	57	137
Maximum vibration displacement Y (μm)							
1	0 (design)	236	138	132	111	27	47
2	200	273	185	174	143	38	74
3	300	292	208	196	159	43	88
4	400	310	232	217	176	49	101
5	500	329	255	239	192	54	115
6	688	364	298	279	222	64	141

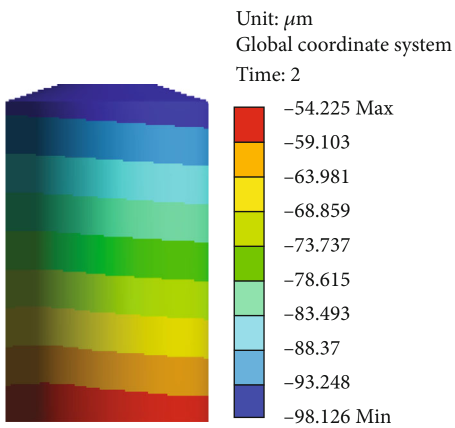
the vibration readings near the motor Non Driving End (NDE) and Motor Driving End (DE) in X and Y direction. X direction represents the direction perpendicular to the discharge flow, Y direction represents the direction along the discharge flow, and A represents the shaft axis. The vibrations are also measured near the pump Driving End (DE) bearings. The measured vibration displacement readings are much higher than the limit of 100 μm as specified by Hydraulic Institute standard (HIS). Vibrations of pumps 1, 4, and 5 are less than the specified limit by HIS.

Table 1 indicates the high magnitude of displacements in pumps 2 and 3. This data is used to verify the impact of vane profile deviation on the performance of a pump using numerical simulations. Figure 2 shows (a) the vibration measurement location, (b) pumps installation at site, (c) manufactured impeller, and (d) scanned data of the impeller, respectively.

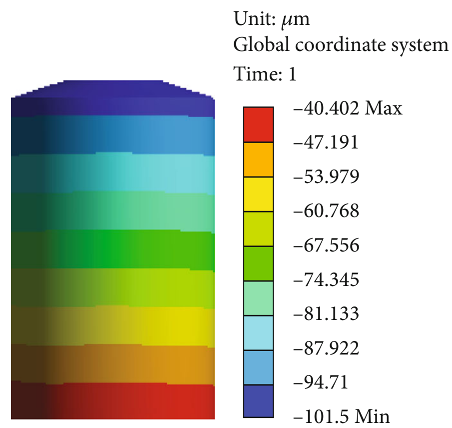
2.7. Numerical Setup Procedure. The stability of any numerical solution depends on the quality of mesh and its robust nature. Since the present approach is a multiphysics solver approach, a careful study of mesh is required in both solvers. Hence, a mesh independence study has been carried out in CFD and FE solvers. The following sections explain the details of the studies conducted in each solver.

2.7.1. Mesh Independence Study (CFD). The unstructured mesh is created using triangular elements for 2D surfaces and tetrahedron elements for 3D flow domain using Ansys ICEM CFD. The hydraulic geometry of pump is divided into four domains: bell mouth, impeller, bowl, and column pipe. 3 layers of prism layers are generated near impeller blade wall surfaces to resolve the boundary layer near rotating wall. The mesh is independently created in fluid domains like impeller, bowl geometry, discharge column pipes, and assembled for analysis purpose. Denser mesh is created near impeller vanes and bowl vane regions. In order to restrict the influence of grid number on the numerical results, a mesh independence study has been conducted with four different sets of mesh. The values of the relative deviation of total head for each mesh with its subsequent mesh for 4 points are shown in Table 2. Mesh 3 is with 1.7% deviation, and mesh 4 is 0.5% deviation with respect to mesh 3. It means that mesh 3 and mesh 4 are producing same consistent results. So, it is decided to consider the mesh 3 as optimized mesh with total node quantity of 0.97 million. Figure 3 shows the plot of total bowl head with respect to mesh numbers.

2.7.2. Mesh Independence Study (FE). The FE mesh is used to subdivide the CAD model into smaller areas called finite elements, based on which a set of equations are solved. These equations approximately present the governing equation of interest with a set of functions which are polynomial defined over each element. As the mesh is refined by making them smaller and smaller, the computed solution will converge to

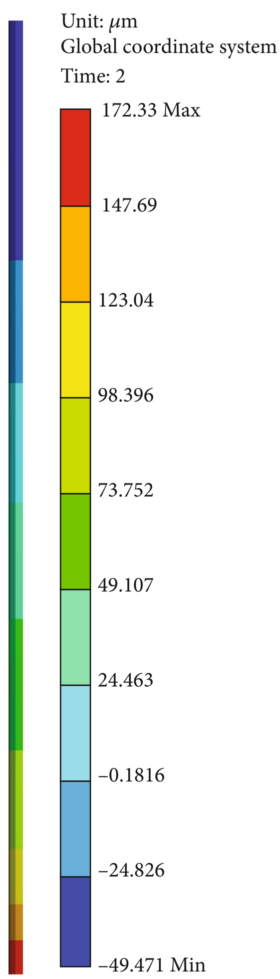


(a) X direction

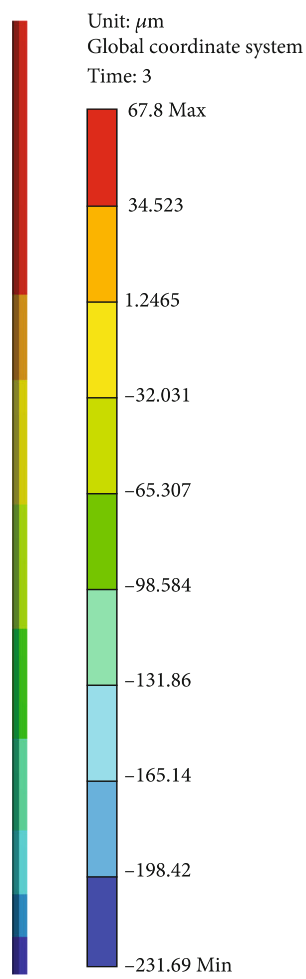


(b) Y direction

FIGURE 6: Displacement contours of motor component in X and Y directions with hydraulic eccentricity of 400 microns.



(a) X direction



(b) Y direction

FIGURE 7: Displacement contours of shaft component in X and Y directions with hydraulic eccentricity of 400 microns.

the true solution. However, computation or simulation time increases as the mesh elements decrease in size. Therefore, an optimum mesh is required to validate any FE simulation through mesh independence.

In this analysis, the mesh independence study is carried out on FE solver using three different mesh sizes to estimate the variation in results such as displacements and stresses. Firstly, a coarse mesh was taken for analysis,

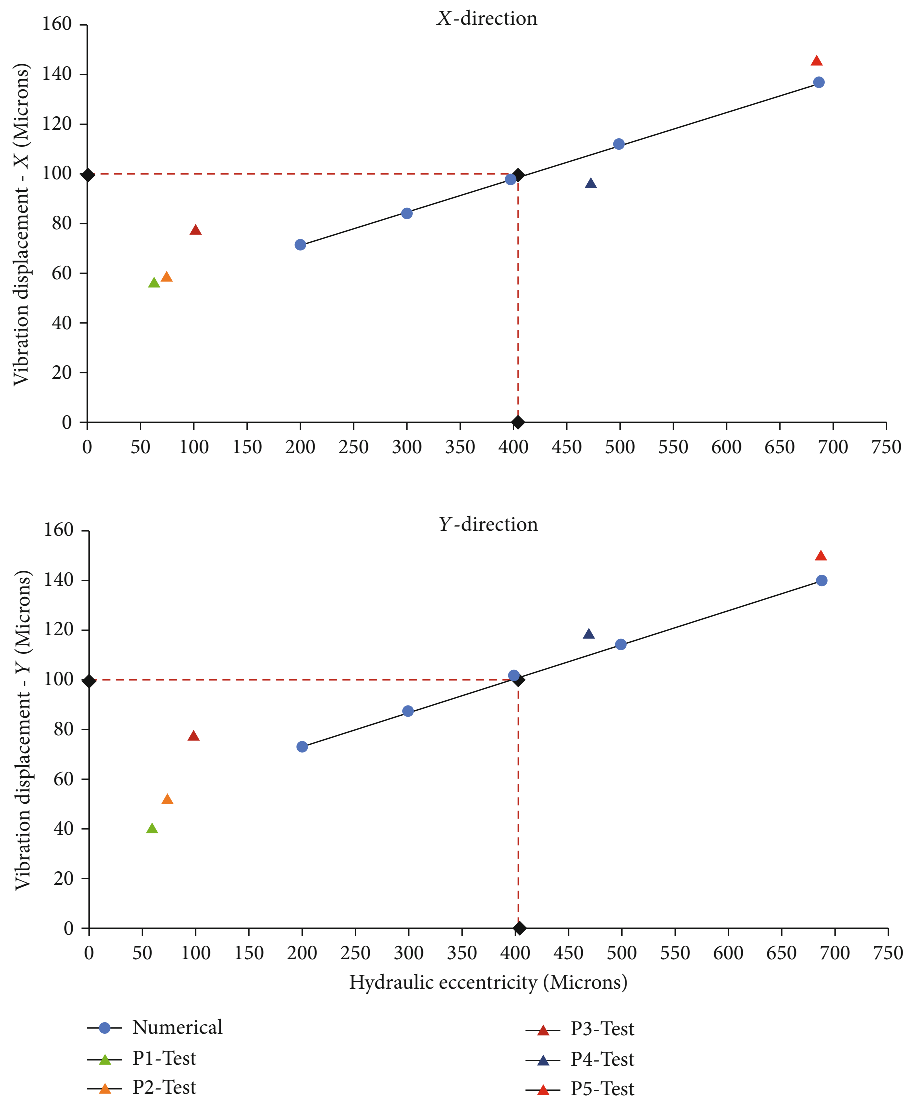


Figure 8: Vibration displacement variation at motor end with hydraulic eccentricity in X and Y directions.

and subsequent finer meshes are taken with same loading and boundary conditions.

The results from mesh independence study are plotted between "Total number of elements" in model (abscissa) vs. "maximum displacements" (ordinate) for all three directions. As seen from Figure 4, the variation between mesh 2 and mesh 3 results is within ±1%. It is observed that the number of elements in mesh 3 is almost 1.5 times (involves higher computation time) as compared to second mesh, but the change in deformation value is less than 1%. Hence, the mesh 2 with 775858 elements is optimum and used in further analysis.

3. Results and Discussion

3.1. One-Way Fluid-Structure Interaction (FSI) Analysis of a Vertical Pump with Maximum Hydraulic Unbalance

3.1.1. Solver Validation. Birajdar et al. [20] conducted one-way FSI numerical simulations on a vertical turbine pump to predict the flow-induced vibration on different components. The process was established for vibration prediction on a VT pump using numerical simulations. The CFD results are predicted initially for total pump hydraulic geometry, and pressure mapping has been done on to the structural model. The pump geometry has been fixed at base plate and discharge flange locations along with the applied hydraulic load as pressure mapping. These boundary conditions are as per the operating philosophy of pump at site. The deviation in the pressure mapping from CFD surfaces to structure part is found to be less than 5%. The output of the simulations is the vibration displacements in X, Y, and Z directions on motor geometry for comparison with the test data. The measured numerical data is reported to be in good agreement with the test data. For a set of assumptions and appropriate boundary conditions, it is proved that the one-way FSI is a good option to predict the flow-induced vibrations in a VT pump. Also, the impact of hydraulic load and hydraulic eccentricity is included in the process. Hence, the FSI methodology is found to be suitable as an initial method with good accuracy to predict the fluid induced vibration on different components in any VT pump and to find the limitation of impeller hydraulic eccentricity and vane pitch. The pressure mapping and the numerical results are explained subsequently.

The process consists of CFD analysis, structural analysis, and comparison with experimental data. The results from the CFD analysis are used as input to the structural model based on the hydraulic eccentricity and unbalance force. Figure 5 shows the pressure contours from CFD and mapped vectors to structural model. Measurements of vibrational displacements near the motor surface are noted and compared with the test data.

Table 3 shows the qualitative data of vibration displacements for both geometries. The displacements in X and Y direction are compared with the displacement of the motor component as per the test data. The deviation that is observed is less than 10%, which can be attributed to the assumptions in the analysis and the one-way application of load.

3.2. One-Way Fluid-Structure Interaction Analysis of Vertical Pumps with Hydraulic Eccentricity as a Variable. Table 4 shows the vibration displacements with respect to hydraulic eccentricity in X and Y directions, respectively. The selected components are the shaft and suction bearing, which are located below the base plate and the motor which is above the base plate. Now, the hydraulic eccentricity value has been varied near the pump impeller in terms of rotational unbalance force. During the experiment, the vibration displacements are measured on the pump motor surface. The maximum limit for vibrational displacement should be below 100 microns. These vibrations are excited due to rotating components below the base plate of motor. The rotating components are rotating with the speed of the impeller, which then leads to nonuniform pressure near the pump impeller. Hence, X and Y directional displacements observed are maximum.

The readings from Table 4 depict that the vibration displacements in both X and Y directions are increasing linearly with the increase in hydraulic eccentricity. Bell mouth, shaft, and the suction bearing are observed with high magnitude of vibration displacements in both X and Y directions as these components are located near to the impeller which has hydraulic eccentricity. Figures 6 and 7 show the contours of the vibrational displacements of the motor and shaft components with hydraulic eccentricity of 400 microns, respectively.

It is observed that, as the hydraulic unbalance increases, the vibration displacements of all component increases in the same manner as of a lever which has a fulcrum in between. Therefore, the vibration displacement value is seen in the increasing order, as one moves away from the fulcrum point. That is, bell mouth vibrations are higher than impeller/shaft, and the impeller shaft vibrations are higher than pump suction bearing vibrations. These components are below base. Motor vibrations are higher than discharge head vibrations which are above base. From Figure 8, for a limiting value of vibrational displacement to 100 microns in X direction, the critical hydraulic eccentricity is found as 405 microns. Similarly, for a limiting value of vibrational displacement to 100 microns in Y direction, the critical hydraulic eccentricity is found as 400 microns. Further, the test data of all 5 pumps is compared with the numerical predicted results. These are found in very close vicinity to the numerical prediction. Using this methodology, the critical eccentricity value is deduced by varying the hydraulic eccentricity of the impeller, to limit the maximum vibration displacement to 100 μm as per HIS standard.

4. Conclusions

In this paper, one-way FSI coupling method is used to determine the interaction between the fluid and structure to determine the vibration displacements of a vertical turbine pump. The pressure mapping has been carried out on the wetted surface of a vertical pump considering the direction of hydraulic load on specific parts. The deviation in the mapping is approximately less than 5%.

The X and Y direction displacements for a maximum hydraulic eccentricity case are compared with the test data and found with a deviation less than 10%. This deviation is mainly attributed to the assumptions and measurements of the readings at site. Further simulations are carried out on the same geometry using hydraulic eccentricity as a variable. The numerical results shown that the components near to the impeller are prone to more vibrational displacements compared to components above the base plate. The allowable vibration displacements as mentioned in the standard are much below to the results obtained from hydraulic eccentricity variation cases. It is necessary to control the hydraulic geometry of the impeller, in turn to control the eccentricity value to limit the lateral vibrations of the pump. For a vertical turbine pump, the critical hydraulic eccentricity is found 400 microns using numerical simulations. It is observed that the one-way coupling fluid-structure interaction method in vertical pumps can be used to understand and predict the fluid-induced vibrations in any vertical turbine pump. Further, it can be used to find and limit the eccentricity of an impeller due to the deviation in vane pitch and hydraulic geometry.

Data Availability

The results of study are new and not found elsewhere.

Conflicts of Interest

The authors declare that there is no conflict of interests regarding the publication of this paper.

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