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ORIGINAL PAPER

Prediction of Flow-Induced Vibrations due to Impeller Hydraulic Unbalance in Vertical Turbine Pumps Using One-Way Fluid–Structure Interaction

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Abstract

Purpose Vertical turbine pumps are used for critical application in power plant. Apart from power plant, these are also used in irrigation, water supply, process industries and petrochemical industries. Centrifugal and vertical turbine pumps consist of rotor geometry and a structure that can vibrate in response to excitation forces. Mass unbalances associated with mechanical and hydraulic geometry of a pump are the two major factors which create dynamic effects in terms of pump vibrations. The generated hydraulic forces resulting due to hydraulic unbalance have similar effect as of mechanical unbalance. For satisfactory operation of the pump, the vibrations in the pumps must be within acceptable limits of applicable standards. Higher level of vibrations not only leads to operational loss, but also leads to down time due to premature failure. Therefore, it is of vital importance for product designers to understand the dominating cause of unbalanced force and its source.

Method The estimation of vibrations using numerical methods can help a designer, especially vibrations arising due to the hydro dynamic forces, for a successful design. In any centrifugal pump, including vertical turbine pump, there is always interaction between fluid and structure. Solid and fluid interaction in present case can be completed by one-way coupling method. The one-way fluid–structure interaction approach is presented in the present paper to predict the vibrations at specific operating conditions which have significant correlation with the test data. Similar philosophy has been applied for an impeller geometry which has hydraulic unbalance to predict the impact of hydraulic unbalance.

Results and Conclusions The benefit of reduction in computational effort and time in this approach can be applied during the initial design stage. Two case studies of one-way FSI approach of a vertical turbine pump are discussed. The approach can be used for evaluating the impact of geometrical deviation, especially vane pitch in an impeller in terms of vibration displacements.

Keywords Pump vibration · Pumping system · One-Way-FSI · Synchronous vibration · Hydraulic unbalance · Eccentricity

Introduction

Vertical pumps are extensively used in wide range of agricultural, municipal, and industrial applications. Vertical turbine-type pump designs are being used for pumping operations in key global industries such as petrochemical, oil and gas, desalination, chemical, power and mining. Few of the applications are, moving processed water in industrial plants, for cooling water circulation in power plants, raw water pumping for irrigation, to boost water pressure in municipal pumping systems. The specific advantages of vertical turbine pump over the other available centrifugal pumps makes them very enviable for most of the applications. Few important advantages of the vertical turbine pump over other pumps are given below:

Self-priming — In this pump, impellers are always submerged in water, eliminating the priming requirement.

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Space savings	Footprint required for a vertical turbine pump is smaller compared to a horizontal pump with electric motor drive.
Suction lift	Vertical turbine pumps must be used where a suction lift is present.

In every working pump, the presence of unsteady forces is inevitable, which can be due to mechanical and hydraulic source. To ensure the safety operation of pumps without affecting the performance, the vibration amplitudes must be kept within specific limits. The causes of vibrations in a pump system can be due to the system or purely related to pump. A distinction between both causes needs to be addressed to diagnose the vibrations in a pump system. The system-related vibrations include unfavourable dynamic response of foundations, piping systems and supporting structures, coupling misalignment, excitations from the driving mechanism and unfavourable approach for flow conditions. Pump-related vibrations include mechanical unbalance of the rotating parts, excessive clearance in seal and bearings causing unfavourable dynamic behaviour, defective bearings and the increased amount of hydraulic forces on rotating impeller. Total pump system vibrations can be further simplified as rotor vibrations, fluid dynamic and structures related to diagnosing a vibration problem. Table 1 [1] shows the details of vibration causes.

Fluid−structure interaction in a pumping system is an important and complex phenomenon which is the prime cause of excessive vibrations in many working pumps. To evaluate the impact of this interaction on any pump or pumping system numerically, the appropriate condition is to evaluate the total system in place of specific component. The presence of unbalance in any rotating component, mainly impeller, affects the flow-induced forces apart from the rotor vibrations. Pump impellers are the core components which are prone to unbalance due to manufacturing and machining conditions at plant. The unbalance in the impeller geometry can be classified as two types: (1) mechanical unbalance, (2) hydraulic unbalance. Mechanical unbalance occurs when there is deviation between mass centreline of a rotating impeller and the centreline of the shaft. Mechanical unbalance is found to be one of the common causes of vibration in which the induced force rotates at $1 \times$RPM, which is also a sinusoidal function of time. Hydraulic unbalance results from (1) uneven flow entry at the impeller, (2) geometrical deviations in impeller. The aforesaid reasons cause a non-uniform pressure distribution on the impeller vane and hence, the unbalanced force. It arises due to the non-uniform blade pitches in a rotating impeller which occurs during manufacturing stage. These unbalanced forces are created from a non-uniform pressure distribution in impeller channels rotating with rotor speed. Hydraulic unbalance increases with increase in flow rate of a pump and dominates mechanical unbalance [2]. Moreover, the magnitude of the unbalance force increases with the increase of the manufacturing deviation [3]. Though mechanical unbalance can be kept in limit by proper balancing of the impeller, there is no direct mechanism available to attenuate the effect of hydraulic unbalance. The causes and troubleshooting of vibrations of a pump depend on several factors, few of them are: the type of pump, type of prime mover, operating conditions, system design, working fluid, impeller design, volute design, upstream flow conditions and conditional monitoring of the total system. Severe vibrations may lead to the failure of shaft which causes operational loss to the customer and sometimes hazardous to the working environment. Cavitation is an important phenomenon in pumps which generates high amount of radial forces and further increases the amount of vibrations [4]. The trouble shooting of vibration problems in pumps at sites is a step-by-step analysis method. Any deviation in the design of impeller causes high levels of sub synchronous vibrations, which lead to cracks in vertical pump shafts [5]. The upstream flow conditions in vertical turbine pump also need to be properly designed to attenuate the swirling action and hence the fluid-induced forces near the bell mouth. In applications of cooling water circulation, vertical pumps are required to have submergence. The non-availability of submergence and adverse upstream flow conditions leads to suction recirculation. Apart from the available sump modifications like back wall splitter, fillets and cruciform, the modifications in impeller throat area and number of blades may reduce the suction recirculation

Table 1 Simplification of pump vibration causes

Rotor vibrations	Fluid-induced vibrations	Structural vibrations
Lateral critical speed of pump	Flow-induced instabilities	Resonances at bearing housing
Torsional critical speed of pump	Flow recirculation in impeller and casing channel	Piping strain causing case distortions
Seal rubbing	Pressure pulsations and acoustical resonances	Resonances of rotating parts, mainly impeller
Unbalance of rotating parts	Flow distribution difficulties in the system	Mechanical resonances due to piping
Coupling misalignment	Water hammer	Resonances at foundation
Instabilities of shaft component	Non-availability of required NPSH	Loose bolting/or grout

[6]. The suction recirculation due to upstream conditions may further lead to the cavitation of the pump impeller. The continuous cavitation of impeller leads to mass erosion from the rotating surface and further fracture of rotating shafts. This is due to the resonance of shaft torsional vibrations, following impeller mass loss due to cavitation. The mass loss further leads to imbalance of the rotating impellers with uneven mass distribution [7]. Theoretical studies for a two-dimensional impeller also showed that the flow unevenness created by a rotating impeller in a pump shows impact on the volute casing [8]. The presence of eccentricity in the flow channels would further enhance the flow-induced forces.

Numerical simulations are widely used for finding the causes of vibration and its possible remedies. One of the basic criteria is to conduct finite-element analysis for a vertical pump to check the resonance. The impact of different possible modifications like structural changes can be conducted and checked easily. By increasing the stiffness of the prime mover base frame, the resonance can be shifted from 1 × operating speed [9, 10, 11]. Numerical simulation methodology was also used for the design of pump intakes, pumps, and pumping systems. This helps in the prediction of performance, flow characteristics, and cavitation phenomenon. The features available in the present CFD tool are effectively used by many industries to check the quality of the flow in a designed pump [12]. Coupling the fluid medium and structure medium in any specific problem is known as fluid−structure interaction, which is a well-known area for predicting the exact behaviour of any system. Prediction of vibrations in any complex piping system using fluid−structure interaction is common area and the numerical results are found to be in good agreement with the test data [13]. Several applications of the fluid−structure phenomenon include the vortex-induced vibration prediction, flow instabilities impact, annular flow-induced vibration and leakage flow instability predictions. In all these applications the resulting interaction of fluid and structure is predicted to well in alignment to the test data [14, 15]. In pumps, the annular flow in seals and its impact on critical speeds and unbalanced rotor system response were predicted using FSI methodology. The impact of L/D ratio in annular seals on critical speeds is predicted and aligns with the test data [16].

The present paper defines a numerical analysis methodology, which analyses vertical turbine pump and predicts the impact of hydraulic unbalance and flow-induced forces on vibration displacements. The methodology is applied for two geometries to compare the impact of hydraulic unbalance on fluid-induced vibrations in a vertical turbine pump. The two geometries are (1) design impeller geometry—in this the source of vibration is flow-induced forces at duty point. (2) Impeller geometry with hydraulic unbalance—in this, the source of vibration is both hydraulic unbalance forces due to shift in axis of rotating fluid volume and the flow-induced forces at duty point.

The methodology consists of fluid flow analysis, pressure mapping on to structure part and structural analysis. CFD analysis of a selected vertical turbine pump has been carried out at duty point using ANSYS-CFX code. The induced pressure on wet surfaces of the pump achieved from CFD is mapped on to the structural part. Structural analysis has been carried out by considering the flow-induced forces and hydraulic eccentricity as the vibration sources. The output of the structural analysis in terms of displacements is measured at motor location and compared with the available test data. The present methodology can be used to study the impact of hydraulic unbalance for specific pump, and the need to limit the hydraulic unbalance of an impeller to limit the vibrations.

Vibrations in Vertical Pumps

Vertical pumps' operation depends on the three factors which are suction conditions of pump intake, piping system on pump discharge side and the system characteristics. The pumping system of a vertical turbine pump is divided into three parts.

(1) Pump intake, (2) pumping system, and (3) discharge system [17].

1. Pump intake or sump defines the flow quality near the pump bell mouth due to upstream flow conditions. Appropriate design of a pump intake/sump is always recommended to avoid any adverse flow conditions such as swirling flow, free surface vortices, and submerged vortices [18, 19].
2. The pumping system of a vertical turbine includes bell mouth and impeller bowl assembly. The presence of vortices or wakes produced in the clearance space between impeller vane tips and the bowl are the major causes of shaft vibrations.
3. The downstream components after the pump bowl are part of discharge system, which are shaft, column pipe, discharge head, and the motor. The adverse effects created by pumping system pass towards the discharge systems. Additionally, the flow output from the pumping system also acts as vibration exciting source.

The foresaid causes of vibrations in vertical turbine pumps are a result of interaction between the pumping fluid and the associated pump structure. The prediction of interacting relationship of fluid and structure and its impact on pump structure in terms of vibrational displacements requires the simulation of complete system.

Fluid–Structure Interaction

Fluid–structure interaction is a kind of multiphysics problem in which the result of related interaction between the structure part and the fluid part can be determined. Any solid structure which has a fluid flow surrounding it is subjected to a pressure induced by the fluid and hence results in structure deformation. The induced deformations in structure cause the change in flow field. The change in flow field further exerts change in pressure on the structural part. This type of repetitive interaction between fluid and structure of a system is known as fluid–structure interaction [20, 21]. Generally, a fluid–structure interaction can be classified into two methods based on the information exchanged (1) one-way coupling, (2) two-way coupling. In case of one-way coupling calculations, only the fluid pressure acting on the hydraulic portion of the structure part is transferred to the structure solver. In case of two-way coupling calculations, the displacements of the structure resulted from the flow-induced forces are also transferred to the fluid solver and the process repeats. The present study is focussed on the one-way coupling method of the fluid and structure.

Numerical Analysis and Vibration Results

Problem Description

The analysis has been carried out for duty point of vertical pump which is, flow (Q) of 30,000 m^3/h, head (H) of 23.5 m at speed of (N) 373 rpm. The numerical analysis has been conducted with the following assumptions.

1. Flow is entering the suction bell uniformly under steady state condition.
2. The baseplate bolts are fixed in all translational directions considering the mounting base as rigid.
3. The piping connection is assumed as rigid due to which the discharge nozzle flange is kept in fixed position.

CFD analysis has been carried out for complete vertical turbine pump at duty point and extracted the pressure data on wet surfaces. The flow-induced pressure data from the wet surfaces of pump are further transferred to structural part which acts as a source of vibration. The vibrational displacements at different locations are the output from the structural analysis, which is due to the implication of source of vibrations. The source of vibrations considered in the present analysis is the flow-induced forces at duty point and the hydraulic unbalance. Hence, two impellers are considered in the present paper in which one impeller is subjected to only flow-induced forces, named as design impeller. The other impeller is subjected to both the flow-induced forces and the hydraulic unbalance force due to eccentricity in centre of mass of fluid volume, which is named as impeller geometry with hydraulic unbalance. Vibration displacements at different locations in three specific directions (X, Y and Z) can be measured from the numerical analysis and can be compared for the design impeller geometry and impeller geometry with hydraulic unbalance. The measured vibration displacements for different components can be used to understand the significance of hydraulic unbalance created by geometrical deviations in impeller vanes. The impact of hydraulic unbalance can be found out using the one-way fluid–structure interaction. The problem has been divided into three parts for both the geometries.

Part 1 CFD analysis of the design impeller geometry and impeller geometry with hydraulic unbalance
Part 2 One-way FSI coupling using pressure mapping
Part 3 Structural analysis of the design geometry and impeller geometry with hydraulic unbalance

The comparison of the design impeller geometry and impeller geometry with hydraulic unbalance is given in the Fig. 1. The mass centre axis of the design impeller hydraulic geometry is collinear with the rotation axis and the mass centre of the impeller with hydraulic unbalance is measured and found to be away from the rotation axis. The measured coordinates were converted to the eccentricity magnitude to quantify the deviation.

The deviation in the impeller can be understood well by geometry overlapping of actual of impeller model over design model. Figure 2 shows the geometry deviation at shroud, vane mid portion, and hub. It can be noticed that the vane pitch between impeller vanes is not uniform. The deviation in the geometry results into the hydraulic deviation with resultant eccentricity of magnitude 688 micron.

Part 1: CFD Analysis

The extracted hydraulic model from the available mechanical model of the vertical turbine pump has been used as input for CFD analysis. ANSYS-CFX (Version 18) code has been used for the fluid flow analysis of hydraulic model and ANSYS-Mechanical (Version 18) code is used for structural analysis, which is best suitable for industrial applications. In the present geometry, the flow near the bell mouth is modelled by assuming a circumferential opening with radius limit of back wall clearance and the height below bell mouth is considered from the available bottom wall clearance at site. Figure 3 shows the vertical turbine pump geometry,

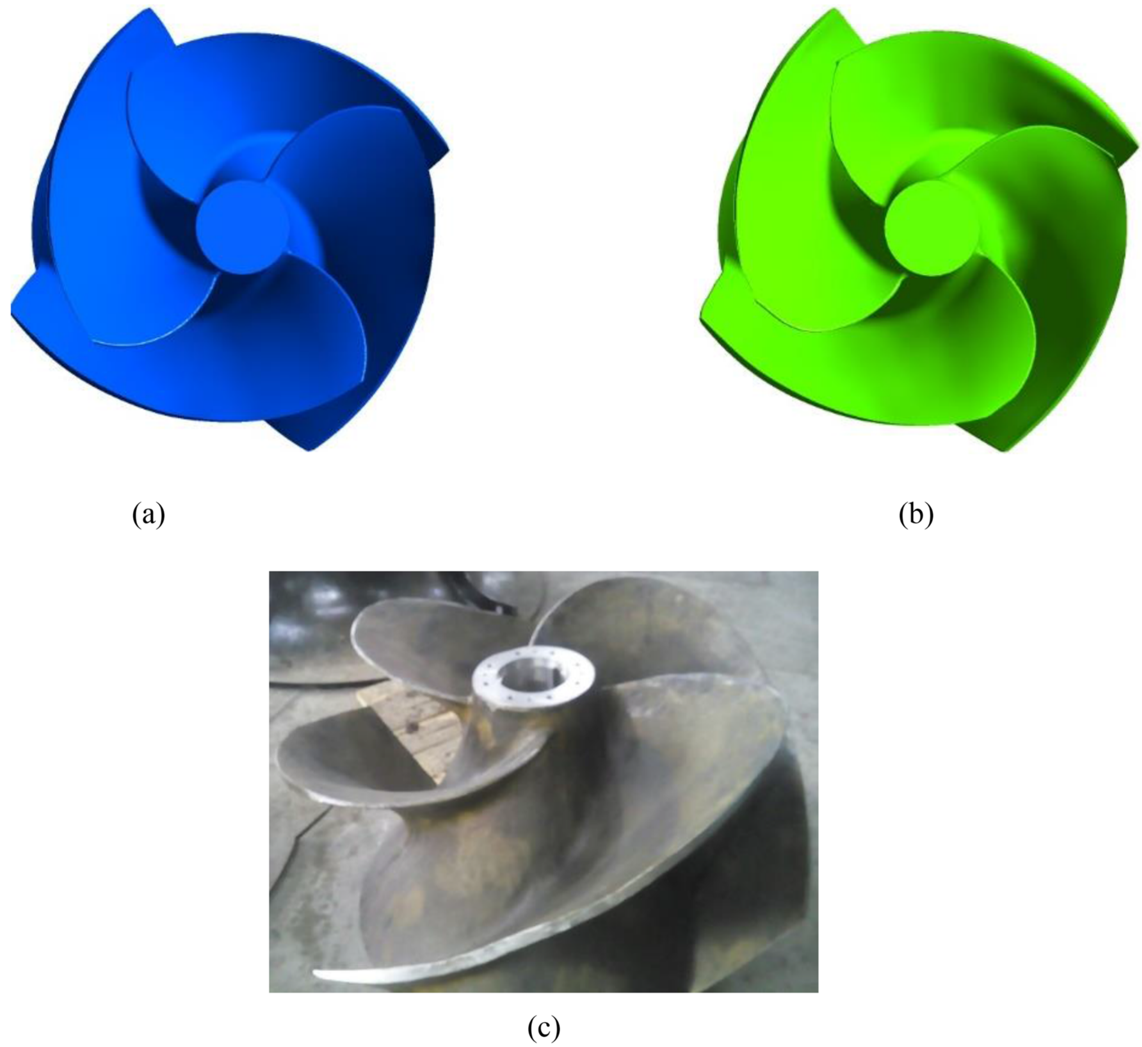


Fig. 1 Comparison of 3-dimensional model **a** design impeller, **b** impeller with hydraulic unbalance, **c** manufactured impeller which consists hydraulic unbalance

which has been used for the analysis. Figure 3a, b shows the pump geometry and section view, respectively. Figure 3c, d shows the extracted fluid volume and the corresponding fluid volume mesh, respectively. Figure 3e shows the finite element mesh of the pump geometry. The duty point flow rate has been specified at discharge outlet and flow opening has been specified on circumferential opening below the bell mouth. The frozen rotor approach has been used for impeller with speed corresponding to duty point. The k-ε turbulence model has been used with scalable wall function, a convergence criterion of 1E − 05 and high-resolution scheme.

Part 2: One-Way FSI Coupling

The numerical simulations and the resulting solutions are based on a partitioned methodology where individual solutions for the different physical fields are prepared. Fluid and structure are the two physical fields in the present analysis. Fluid field has been solved by CFD methods using ANSYS-CFX code and the structural part has been solved by structure dynamics methods using ANSYS-Mechanical code. In one-way FSI coupling, the resulting pressure values on the internal walls of the hydraulic model have been mapped on to the structural model, which acts as a source of vibration. Figure 4 shows the pressure contours on hydraulic part of the pump and the mapped data in structural model. In ANSYS pressure mapping method, the absolute pressure values (negative to positive) from fluid model are converted to zero-based values in structural model, i.e., positive pressure values are mapped as it is, but the negative pressure values are converted to positive values with reversed normal direction. Hence, no negative pressure values appear in the structural model of the pump after pressure mapping as shown in Fig. 4. The

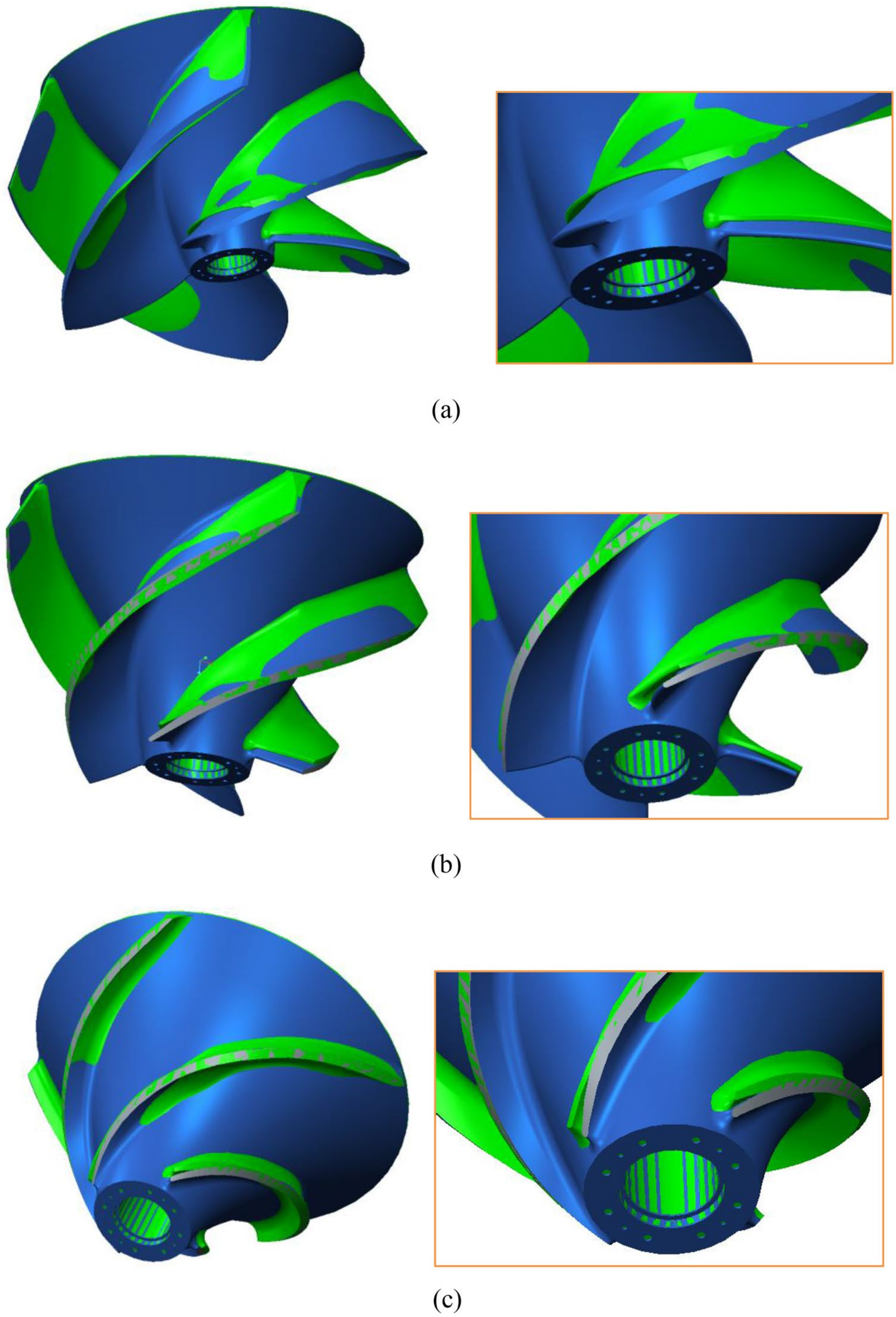


Fig. 2 Comparison of impeller model at different sections **a** near shroud, **b** near vane mid-section **c** near hub. Blue colour indicates design impeller and green colour represents impeller with hydraulic unbalance

deviation found in mapping pressure data is found to be only 5%.

Part 3: Structural Analysis

The internal wet surfaces of the pump and the structural model of the motor geometry are the driving dimensions for shell modelling to attain pressure mapping from the wetted parts with least possible deviation. The mass quantity of the fluid which is displaced due to the pump submergence has been added along with mass quantity of the fluid portion in total vertical turbine pump. As mentioned in assumptions, the discharge nozzle has been fixed at flange location along with the foundation bolt pads. The pressure values resulted from CFD have been mapped on to structural part, which is considered as applied structural loads.

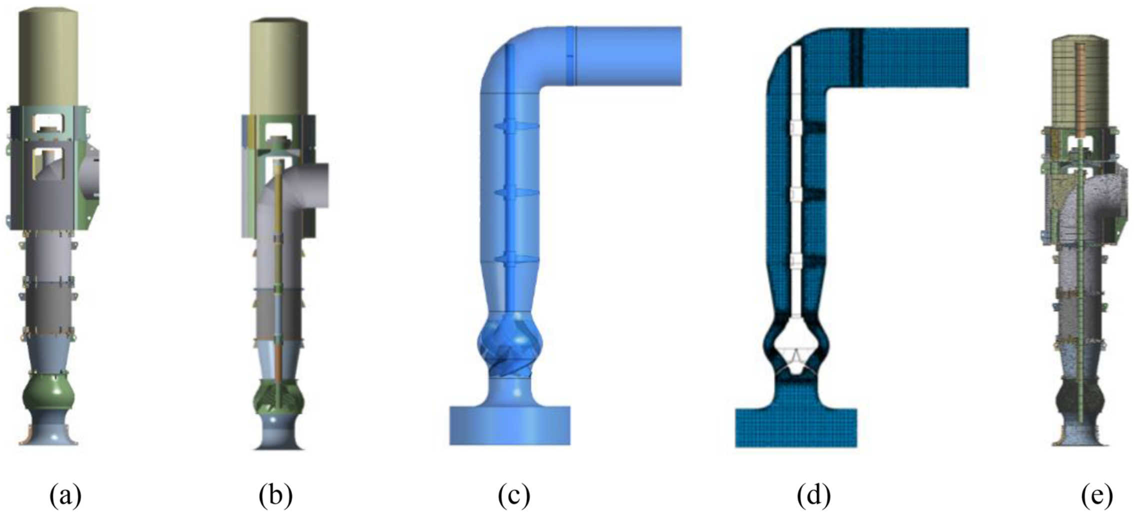


Fig. 3 Vertical turbine pump geometry and mesh at different stages of analysis. **a**Vertical pump geometry, **b** section view, **c** fluid volume, **d** finite volume mesh of fluid volume, **e** finite-element mesh of structural part

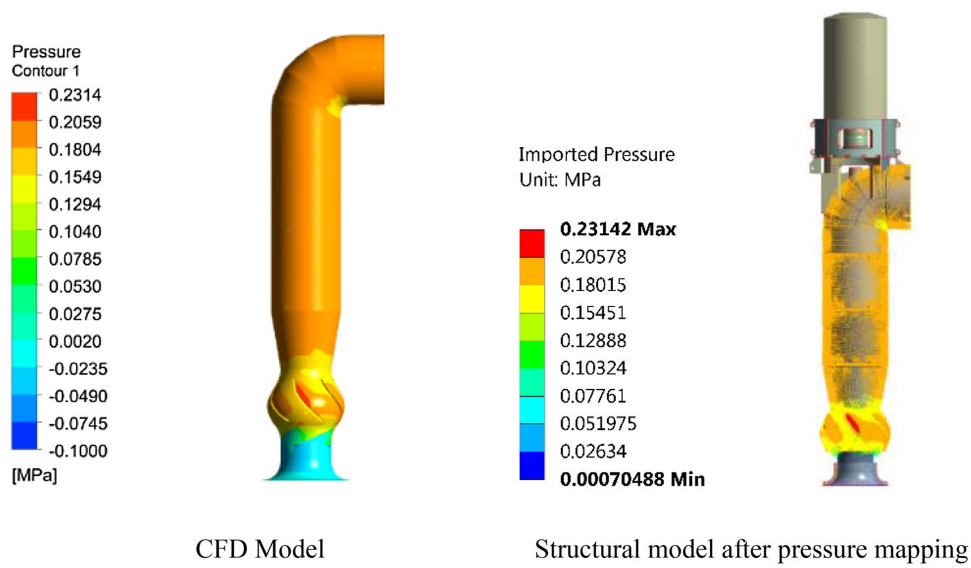


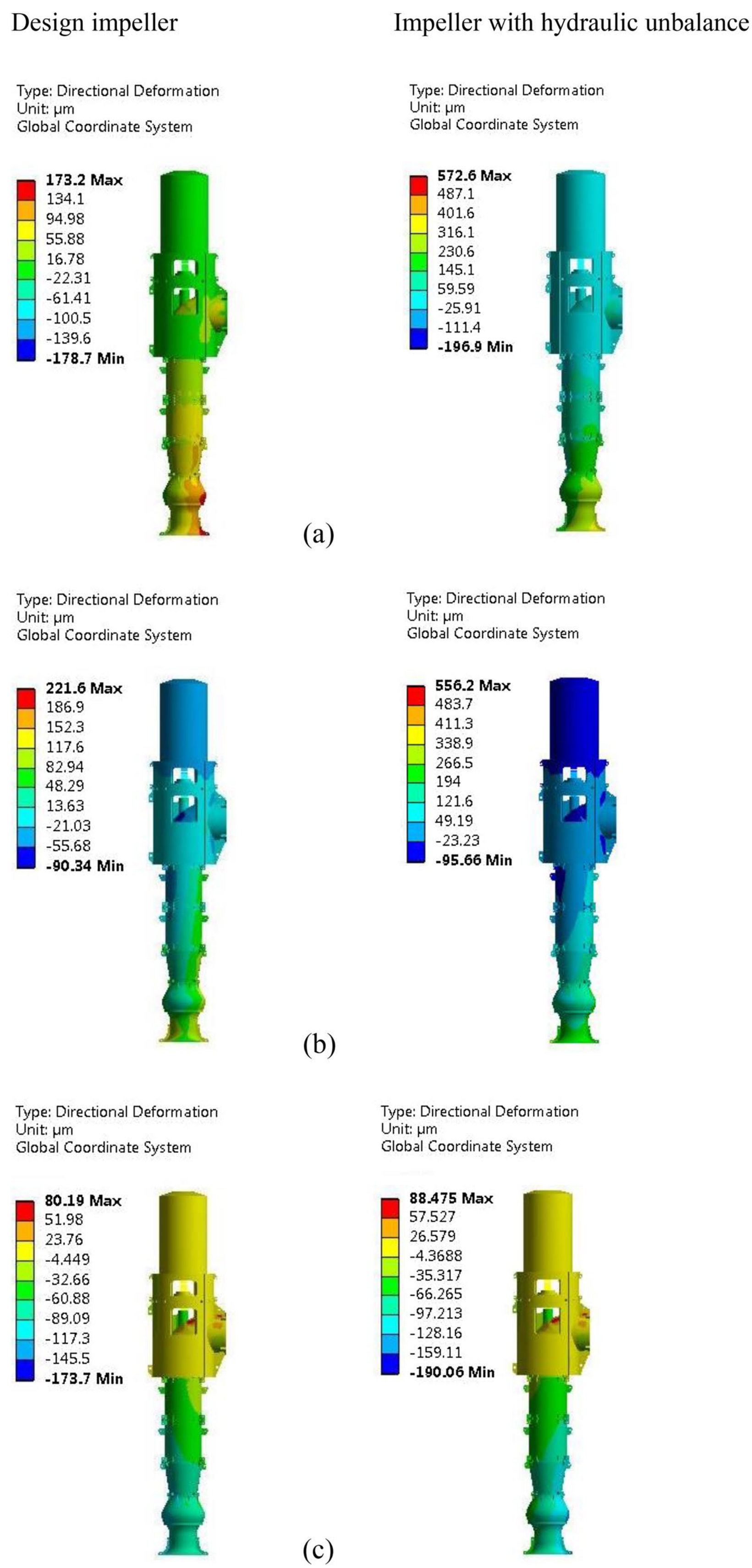
Fig. 4 Pressure mapping from CFD model to structural model

Numerical Results

Numerical results of both the geometries are compared qualitatively and quantitatively. The qualitative results comparison is given in Figs. 5 and 6 in terms of vibration displacement contours of total pump system and motor, respectively. The quantitative results are given in terms of vibration displacements in the three directions for few major components. Table 2 shows the vibrational displacements' comparison of tested data and numerical results with design impeller. Table 3 shows the vibrational displacements' comparison of test data and numerical results of impeller geometry with hydraulic unbalance. The available test data in both the tables consist of vibration displacements in microns measured at the top of the motor in three directions *X*, *Y*, and *Z*. *X* denotes perpendicular direction to discharge, *Y* denotes along discharge direction, and *Z* denotes along the shaft axis. Figure 5 shows the comparison of lateral vibrations in both the geometries. It is also found that the impeller geometry with hydraulic unbalance shows the maximum displacements in lateral directions, which are due to the unbalance created by the impeller hydraulic geometry. The displacement contours are also given for the motor geometry of both the geometries. From Fig. 6, it is understood that the motor with hydraulic unbalance impeller geometry shows slight increase in the lateral vibrations.

Tables 2 and 3 show the quantitative data of vibration displacements obtained from numerical simulations for both the

Fig. 5 Displacement contours on vertical turbine pump in three directions **a** *X*, **b** *Y*, **c** *Z*



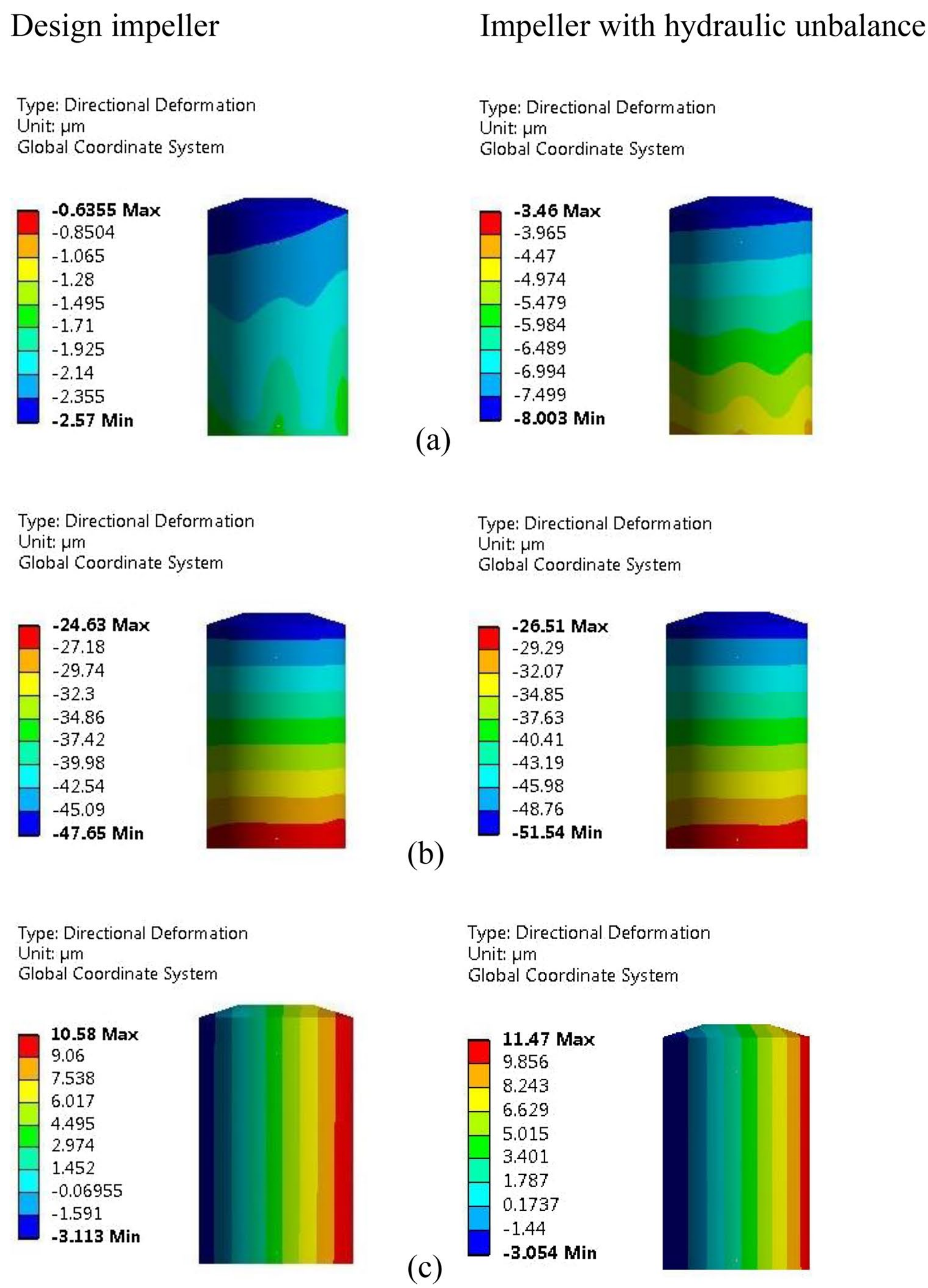


Fig. 6 Displacement contours on pump motor in three directions **a** *X*, **b** *Y*, **c** *Z*

Table 2 Vibration displacements in design impeller

Component	Vibration displacements in microns (µm)					
	Test data (µm)			Numerical results (µm)		
	X	*Y*	*Z*	*X*	*Y*	*Z*
Shaft	–	–	–	74	116	2.1
Suction bearing	–	–	–	64	102.3	2.7
Motor	64	62	15	2.57	49	11
Discharge bend	–	–	–	178.7	90	80
Bellmouth	–	–	–	167	221.5	165

Table 3 Vibration displacements in impeller with hydraulic unbalance

Component	Vibration displacements in microns (μm)					
	Test data (μm)			Numerical results (μm)		
	X	*Y*	*Z*	*X*	*Y*	*Z*
Shaft	–	–	–	572	556	3.1
Suction bearing	–	–	–	274	260	3.5
Motor	–	–	–	8	51	11.5
Discharge bend	–	–	–	201	94	91
Bellmouth	–	–	–	382	360	182

geometries. The numerical results show that the vibrational displacements in *Y* and *Z* direction of motor component are found to be in good agreement with test values at initial guess. The source of vibration which is the fluid-induced forces are resulting vibrations in the direction of discharge, which is *Y*, are well predicted using one-way FSI approach. The data show that the major impact of the hydraulic unbalance forces occurs on shaft following impeller suction bearing. The displacements on the motor are found to be slightly increased due to the hydraulic unbalance. It shows that the vibration transfer from shaft to motor is found to be minimum. This could be due to the assumption that the mounting base is considered as rigid (infinite stiffness of base), which is an ideal condition. However, as the clearance between the suction bearing and other line shaft bearing increases over period of time, the vibrations at motor level also increase.

The vibration displacements mentioned in Table 2 are for design impeller which is only due to fluid-induced forces. However, Table 3 mentions vibration displacement for an impeller with hydraulic unbalance along with fluid-induced forces. It is further noted in Table 2 that the displacement in stationary parts (i.e., discharge head and bell mouth) is found to be relatively higher than the rotating parts. The flow from circumferential inlet causes the non-uniform pressure distribution on bell mouth and the resulting flow from bell mouth also causes non-uniform pressure distribution on discharge bend, and hence higher the flow-induced force on respective components. But under given assumptions and boundary conditions for design impeller case, in shafts, the impact of the flow-induced forces is lower compared to bell mouth. However, under similar conditions, from Table 3 it is observed that the rotating parts are showing higher vibration displacements, which are due to dominating effect of hydraulic unbalance over the flow-induced forces.

The components are numbered with preference to compare the absolute maximum displacements of both the geometries with graphical representation. Shaft, impeller suction bearing and motor are named as component 1, component 2 and component 3, respectively. The hydraulic unbalance in the geometry is causing the maximum lateral vibrations in the shaft component with a range of 556−572 microns. Figure 7 shows the comparison of vibration displacements of shaft component for both the cases. Following the maximum displacements in shaft, impeller suction bearing is found to be with lateral vibration displacements ranging 260−274 microns. Figure 8 shows the location of impeller node and suction bearing node. The impeller is modelled as lumped mass and bearings are modelled as spring elements. Vibration displacement is plotted for impeller node and the suction bearing. Figures 9 and 10 show the same with comparison of two cases. From Table 3 and the contours, it is clear that the displacements are increased compared to the measurements in design impeller. This is due to the major fact of hydraulic unbalance in the impeller geometry which is acting as a source of excitation. The effect of hydraulic unbalance in impeller geometry is causing the unbalance force on shaft which corresponds to 1 × excitation frequency as observed in several literature (Fig. 11).

Conclusions

In the present paper, the interaction between the fluid domain and structure domain of a vertical turbine pump has been simulated numerically using one-way coupling method to predict the flow-induced vibrations on different components. The pressure mapping has been done on the wetted surfaces of a selected vertical pump with a precise care about the hydraulic load direction on specific parts. The vertical pump has been fixed at discharge flange end and near the base plate in numerical model as a boundary condition which aligns with the site conditions. The deviation in the pressure mapping from hydraulic part to structure part is found to be less than 5%.

Based on available test data, the vibrational displacements achieved from the numerical simulations are found to be in good agreement with test values. Vibration displacements are measured near motor location for directions *X*, *Y*, and *Z*, compared with the respective test values. Same philosophy is applied for a case with impeller geometrical deviation. The deviation in impeller vane pitch geometry is causing the deviation in hydraulic geometry

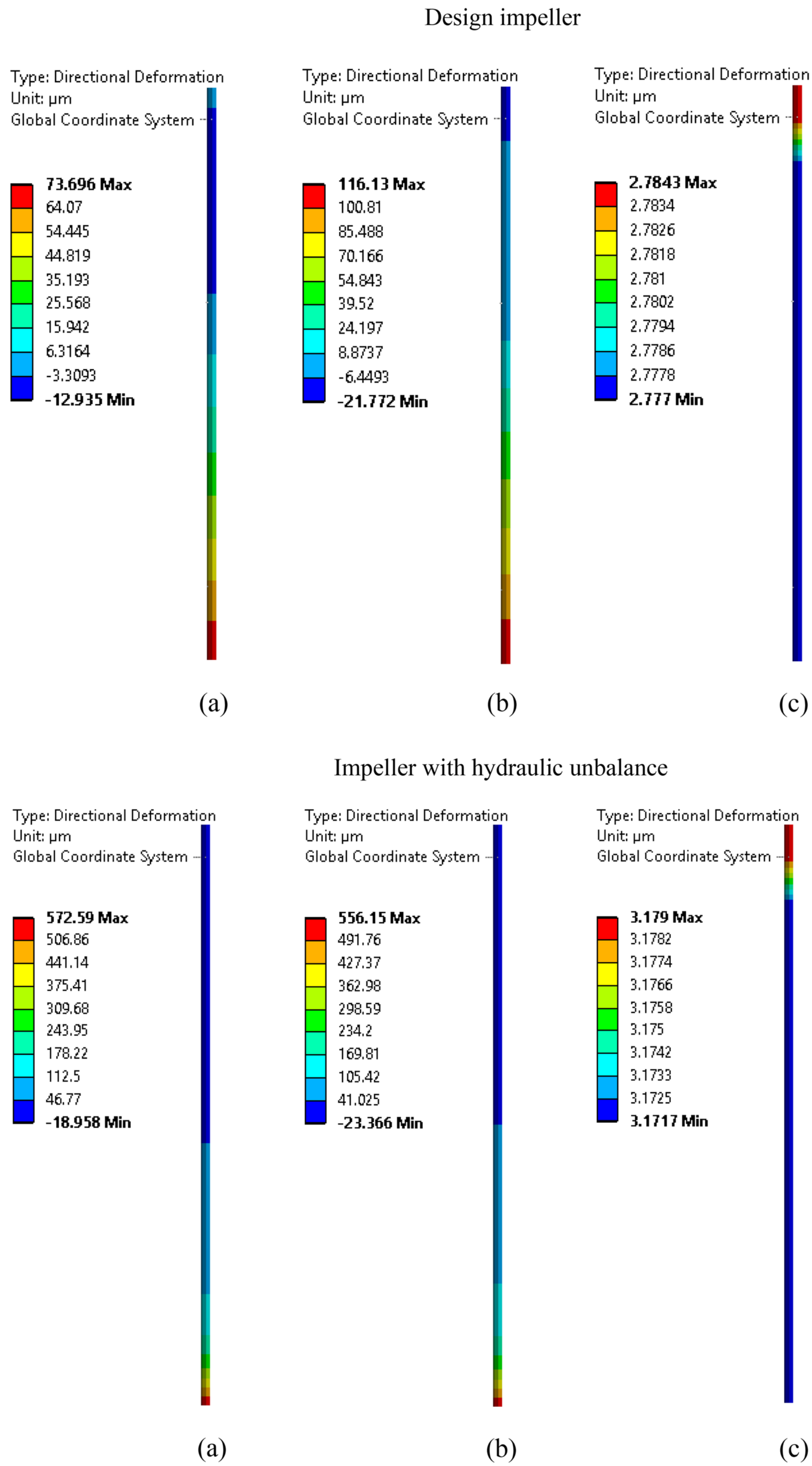


Fig. 7 Displacement contours on pump shaft in three directions **a** *X*, **b** *Y*, **c** *Z*

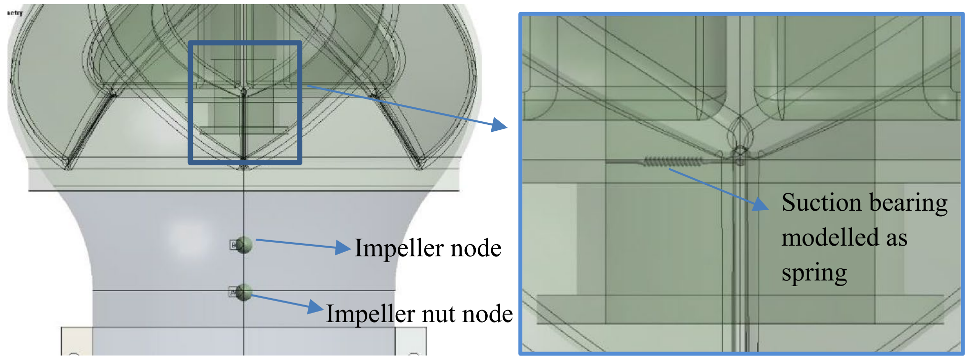


Fig. 8 Locations of impeller and suction bearing

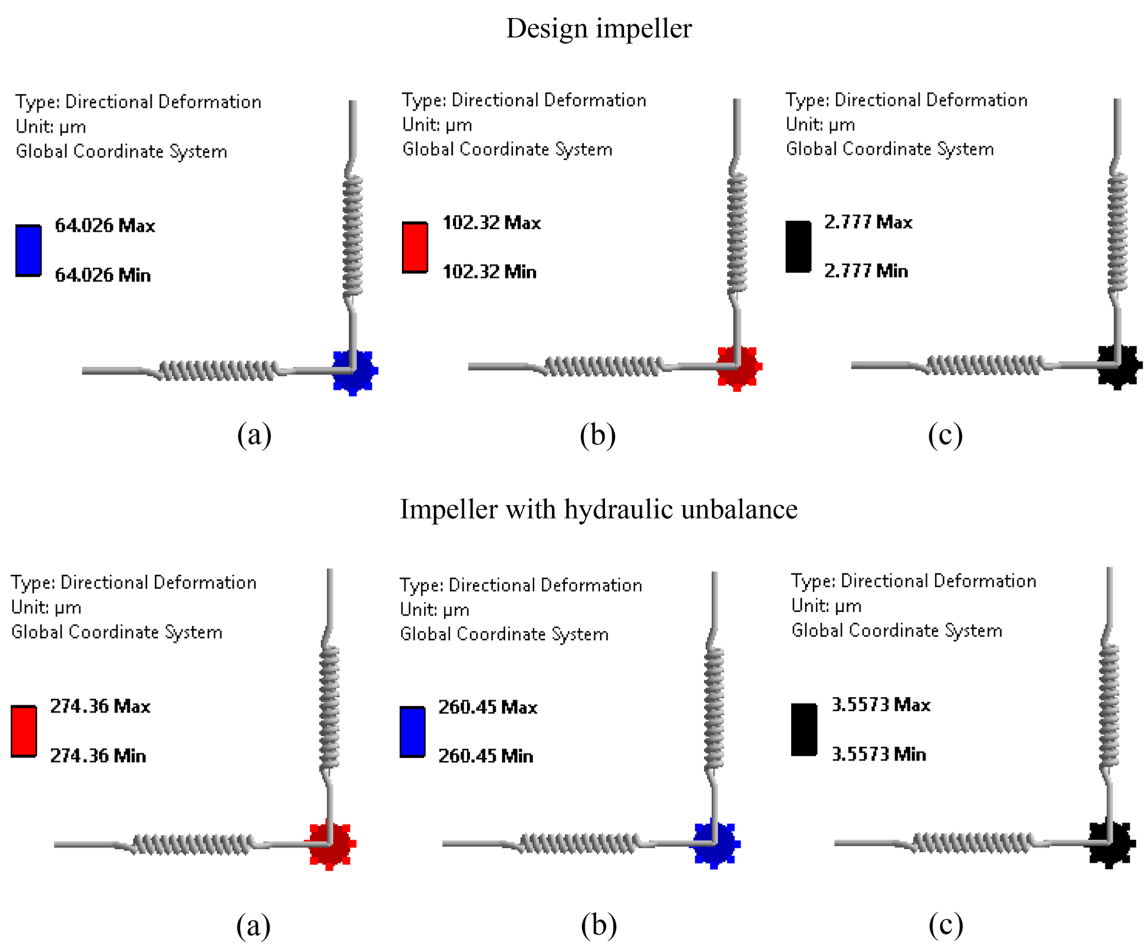


Fig. 9 Displacement contours on suction bearing in three directions **a** *X*, **b** *Y*, **c** *Z*

of the vanes, and hence, hydraulic unbalance. Absolute value of maximum displacement is observed to occur in shaft component following the impeller suction bearing. Eccentricity of the impeller hydraulic geometry is the prime parameter which affects the vibration displacements. It is necessary to keep tolerance on eccentricity value for an impeller hydraulic geometry, especially vane pitch to limit the lateral vibrations in a pump unit. Overall, the one-way coupling fluid–structure interaction method in vertical pumps predicts the vibrational displacements using source of vibrations as flow-induced forces and hydraulic eccentricity. It can be used as an initial method with good accuracy to predict the fluid-induced vibrations on different components in any vertical pump and to find the limitation of impeller hydraulic eccentricity and vane pitch.

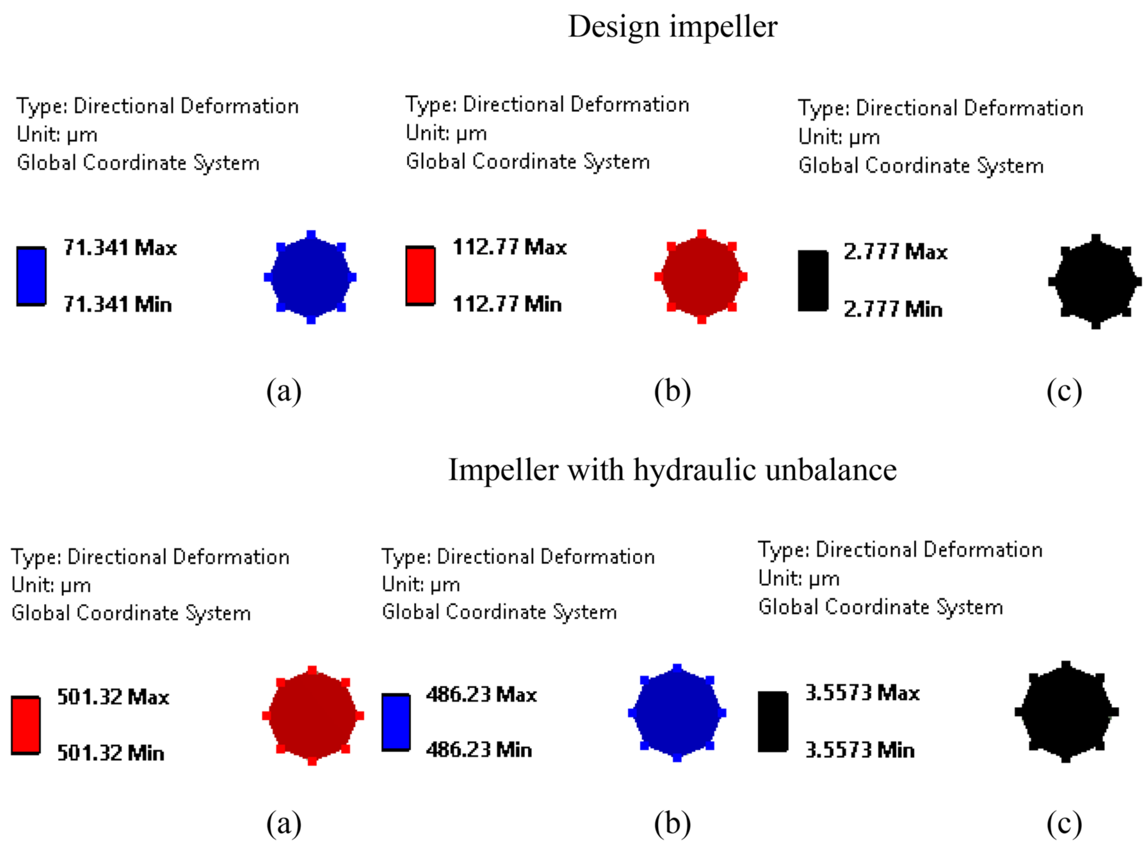


Fig. 10 Displacement contours on impeller node in three directions **a** *X*, **b** *Y*, **c** *Z*

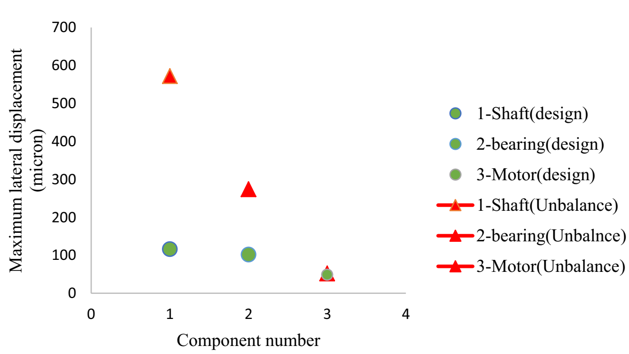


Fig. 11 Graphical representation of lateral displacements in design impeller geometry and impeller with hydraulic unbalance

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