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# Dynamic Stability of Zaghloul Drainage Pumping Station, Kafr El Shiekh, Egypt

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**ABSTRACT**
Zaghloul Pumping Station located at El-Moheet drainage in Kafr El Sheikh Governorate. The station consists of four axial pumping units. The operating system depends on running not more than three pumping units. The old station is replaced by new one and the old sump is reused with some modifications. Each pump takes 20 second to start and takes 300 second to reach steady state. Activation more than two units lead to decrease the suction water level, disturbance of velocity distribution in suction intake and increase vibration and noise levels. The disturbance in velocity distribution generates dynamic instability of pumping units which leads to failure, damage and other operation and maintenance difficulties. Field measurements and numerical simulation were done to investigate dynamic stability of the station. The Solid Works flow simulation software, Computational Fluid Dynamics (CFD) is used to simulate the flow conditions at different water levels to predict the hydraulic problem at the suction side. The field measurements are used to investigate the dynamic problem. Measurements are used to measure pump flow rate, pump head and vibration levels. The (MVP2C OneproD/ ACOEM) two channel vibration analyzer and data collector was used to prepare dynamic balancing for each unit with different weights. In general, the results indicated that with the decrease of water level; approach velocity increases, swirl and vortices induce vibrations and excessive bearing loads. From simulation results the geometry of suction intake is proper for running three parallel axial flow pumps with the designed flow rate. The input data to simulation model were taken from the filed measurements and this data is validating to simulation model. The dynamic balancing as a solution leads to reduce vibration level and save bearing life. It is very important to investigate the dynamic stability and check the dynamic balancing for pumping units at the primary operation of the new stations as mentioned in the bidding.
***Keywords*** - Dynamic stability; Vibration level; CFD; Velocity distribution; Intake design

## I. INTRODUCTION

In Egypt, there are more than 1,500 irrigation and drainage pumping station. These pumping stations are exposed to many problems such as hydraulic, dynamic and electrical ones, which affect the performance, safety and efficiency of these stations. These problems are costing the country millions of pounds of foreign currency for operation and establishment, maintenance and purchase of spare parts for these stations to maintain the distribution of water in Egypt [1]. They require a continuous assessment, surveillance, monitoring and analyzing the performance and avoid the problems that accelerate the collapse and damage the component of these pumping stations. It is an accepted fact that faulty design of pump sump or intake is one of the major causes of unsatisfactory operation of pumps in any pumping station, Desmukh and Gahlot [2]. The flow conditions at entry to a pump depend upon flow conditions in approach channel, sump geometry, location of pump intake with respect to the walls, velocity changes and obstructions such as piers, screens etc., and rotational tendencies in flow produced upstream of the pump bays [2]. A number of deficiencies and problems with vertical pumps are often related to sump design rather than mechanical imperfections. When a pump intake bays are not properly designed, severe swirling flow problems may occur in the pump bay, Mohd Remy et al [3]. One of the sources of disturbance at intakes is the existence of free-surface vortices with an air core. The most common solution for avoiding air-entrainment is the use of anti-vortex devices and, especially, plates for large pipe or shaft intakes, Borghri and Kabiri [4]. The flow characteristics of pump sump and performance analysis of the mixed flow pump should be done. The efficiency of anti-vortex devices to insure the uniform flow is confirmed. From the numerical analysis, the inception of cavitation is observed on the suction surface where the leading edges meet the tip, and then the cavitation zone expands, Y X Zaho et al [5]. Numerical analysis for the flow characteristics of a sump model with pump intake and good agreements were achieved by comparing the numerical results with the experiments, Rajendran et al [6]. Submerged vortex is introduced numerically by analyzing the flow in the pump sump with and without baffle plates, Iwano et al [7]. A multi-intake pump sump model analyzed by using CFD analysis to check the flow uniformity by predicting the location, number and vorticity of the vortex, Lee et al [8]. A design guideline for the shape of pump sump is suggested by model test and CFD analysis, Lee [9]. Moreover, Turbo-machinery Society of Japan [10] has revised the standard of pump sump model test and the revised standard examined the possibility of numerical analysis for the prediction of the flow in the sump model using several commercial and in-house CFD codes. The flow uniformity according to the flow distribution in the pump intake channel is examine to find out the cause of vortex occurrence in detail by experiment and CFD, Kim et al [11]. Quantitative measures of axial and tangential velocities and swirl angles were made in the physical hydraulic model using a 2-D laser Doppler velocimeter (2-D LDV) and traditional swirl meters. The results indicate that additional research is needed to develop application guidelines and acceptance criteria for use of CFD models in evaluating flow entering pump inlets, Joseph Orlines et al [12].

## II. PROBLEM IDENTIFICATION

Zaghloul Pumping Station is located at El-Moheet drainage in Kafr El Shiekh Governorate. The station consists of four axial pumping units as shown in Figure (1). The pump operating point is 6 $m^3/s$ discharge and 4.9 m total head. The pumps driven by 450 Kw, electric motor with gear box for each unit. The pumping station suction level varied from -3 m to -2.2 m. The discharge water level varied from 1 m to zero m. After installation and operation of two units, the pumping station is exposed to high vibration and noise levels. Activation more than two units lead to decrease the suction water level, disturbance of velocity distribution in suction intake and increase vibration and noise levels. The disturbance in velocity distribution generates dynamic instability of pumping units which may leads to failure, damage and other operation and maintenance difficulties. The numerical simulation is used to evaluate the velocity distribution, pressure distribution and contour of velocity in sump of Zaghloul Pumping Station. The field measurements were used to monitor the hydraulic and dynamic stability of pumping units. Therefore, the objective of this work was to use numerical and experimental studies on dynamic stability to assess the current problems in Zaghloul Pumping Station. The research also provides recommendations for safe operation conditions at two suction water levels.

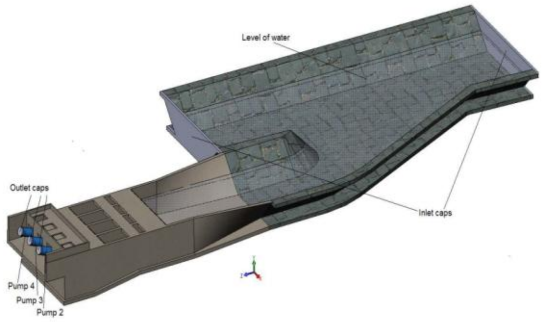



Figure 1: General layout sump of axial flow pump and intake configuration

## III. EXECUTING SIMULATIONS

The basic three-dimensional geometry is prepared using Solid Works flow simulation software drawing with the grid surfaces plot of sump. This model is one of the best theoretical programs used in solving flow problems in pumping station intakes. Figure (1) shows the overall pump intake configuration of the pump intake model simulated in this study. In this figure, the water inlet from two directions, and exist through four water caps inlet to four vertical pumps. On the other hand side water level in pump suction intake varied from -2.2 m to -3 m. The first simulation scenario of model discharge,($Q_m$) is 6.62 $m^3/s$ and suction water level,(WL) is -2.2 m. The second simulation scenario of model discharge,($Q_m$) is 6.00 $m^3/s$ when suction water level,(WL) is -3 m.

### 3.1 Pump Initiation

In this research were selected two discharge values based on field measurements. The numerical analysis is transient state. Each pump takes 20 seconds to activate, with a linear ramp from 0 to maximum flow rate. Pump No. 2 is activated at the start of the simulation, pump No.3 after 5 minutes and finally pump No. 4 after 10 minutes since beginning of calculation. Simulation ends after 15 minutes. Pump No. 1 remains inactive for the whole process.

Therefore, the sequence of events are as follows: 0 second start of simulation and beginning of activation of pump No. 2, 20 second end of activation of pump No. 2 , which now works at nominal flow rate, 300 second beginning of activation of pump No. 3, 320 second end of activation of pump No. 3, which now works at nominal flow rate together with pump No. 2, 600 second beginning of activation of pump No. 4, 620 second end of activation of pump No. 4, which now works at nominal flow rate together with pumps No. 2 and No. 3, and 900 second end of simulation. The "ideal wall" condition simulates the level of water, while the environment pressure condition is applied to simulate on the "inlet caps".

## IV. GEOMETRY AND GRID GENERATION

The quality of the computational mesh has an important role in achieving the desired accuracy of the simulations especially if the computational domain is very complex. The Solid Works flow simulation software drawing with the grid surfaces plot of sump. The equations are supplemented by fluid state equations defining the nature of the fluid, and by empirical dependencies of fluid density, viscosity and thermal conductivity on temperature. To predict turbulent flows, the Favre-averaged Navier-Stokes equations are used, where time-averaged effects of the flow turbulence on the flow parameters are considered, where the other, i.e. large-scale, time-dependent phenomena are taken into account directly. Three dimensional unstructured meshes are used for the flow simulation in the pump sump. Unstructured mesh is used for this study due to model complexity and easy to accurate solve the governing equations of the model at intake section. The numerical solver uses unstructured meshes which allow flexibility in meshing very complex geometries while maintaining high quality computational mesh which is necessary for obtaining accurate solutions [13]. The three-dimensional unstructured mesh (tetrahedral) has been created by using the structured mesh (hexagonal) as shown in Figure (2). The number of cell in X-direction is 20, the number of cell in Y-direction is 4 and the number of cell in Z- direction is 64. The total number of mesh elements is 94382. After the meshing is completed, the boundary condition was set through a command zone provided by pre-processor. The 'Environmental pressure' was set at inlet while 'volume flow' for the outlet region. The boundary conditions is water surface set as 'free surface condition with a constant air pressure' boundary and pump model is set as 'walls' including the suction pipes as shown in Figure (3).

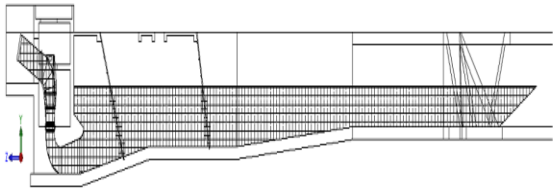


Figure 2: Computational grid

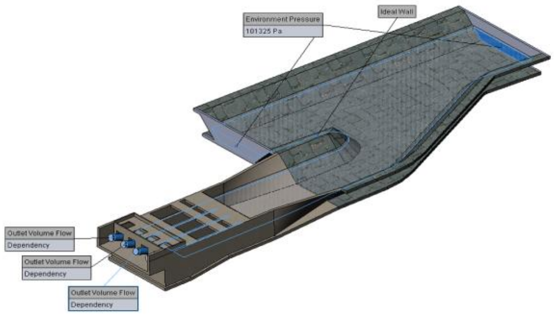



Figure 3: Simplified model boundary conditions

## V. RESULTS AND ANALYSIS

Results were obtained, analyzed and presented, as follows.

### 5.1 Field Measurements

Test facilities and Instrumentations are very important to evaluate the performance of any pumping station. Generally, performance curves for pumps developed by the pump manufacturer are based on tests of a single pump. Flow and pressure measurements and vibration levels are required for performance analysis. A transit-time ultrasonic flow-meter type (1010) was used to measure the volume flow rate through the pipeline. It is the best technology to measure the flow-rate through pipeline from its outside without any damage or leakage [14]. A calibrated pressure transducer was used to measure pressure at the delivery side of the pumping units. Forced vibration test was done to determine vibration levels and exciting frequencies using (MVP2C OneproD/ ACOEM) two channel vibration analyzer and Data collector with serial 34234. A machine monitoring software package, (XPR300 version 5 Premium editions) was used for evaluating dynamic running condition of the pumping unit.

**5.1.1 Hydraulic Performance**

Hydraulic performance tests are done during normal operating conditions around the design point, where the control valve in the delivery pipe is used to control flow rate. Discharge, suction level, and delivery pressure are monitored and total head is computed for unit No.2 as example. The pump operating point is 6 $m^3/s$ discharge and 4.9 m total head. There are two actual operating points measured at two different water levels at 305 rpm measured pump speed. The first one is 6.62 $m^3/s$ at suction water level of -2.2 m. The second flow rate is 6.00 $m^3/s$ at suction water level of -3 m. Consequently the measured operating points are located at the manufacture (H-Q) curve as shown in Figure (4). On the other hand side when start pump number two there are excess vibration and hydraulic problems appears. However, the flow details, which could not be experimentally obtained inside the pump, are numerically obtained.

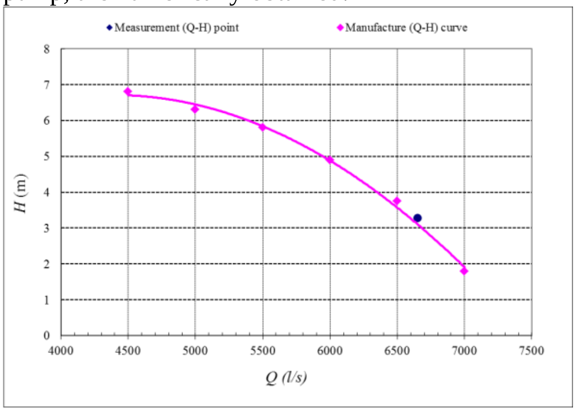



Figure 4: Manufacture (Q-H) curve verses measurement (Q-H) point for unit No.2

**5.2 Numerical Simulation Results**

The input data to simulation model were taken from the filed measurements and this data is validating to simulation model. The validation of simulation model was done to insure that model is properly simulating the field measurement. Different scenarios of a pump intake are simulated using CFD

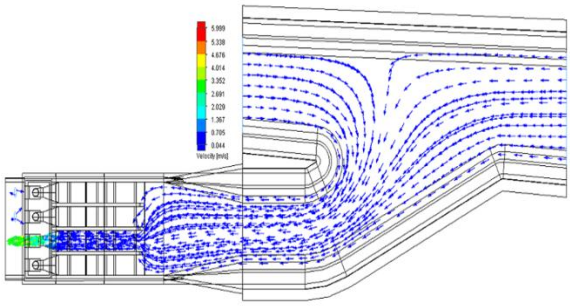
to find the effect of suction velocity distribution at intake when reducing suction water levels under design water level. Through this different scenarios the vibration level of pumping station were measured to insure that dynamic stability of pumping station.

**5.2.1 Velocity distribution at pump flow rate equal to 6.00 $m^3/s$ and suction water level of -3 m,**

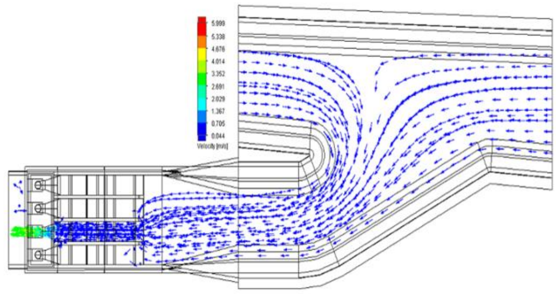
The velocity distribution at the suction sump shows that the flow was symmetrically directed towards the pump intake and no subsurface vortices are presented in this region as shown in Figure (5). In this case, simulation is conducted with three pumps are running in parallel at the same time. The flow reached to all running pumps and there is no circulation zones appeared at the pump sump. This means that the design of the exist sump by the exist dimension is suitable for running three parallel axial flow pumps with flow rate equal to 6.00 $m^3/s$ for each pump at suction water level of -3 m.

**5.2.2 Relative pressure plots at pump flow rate equal to 6.00 m3/s &suction water level of -3 m**

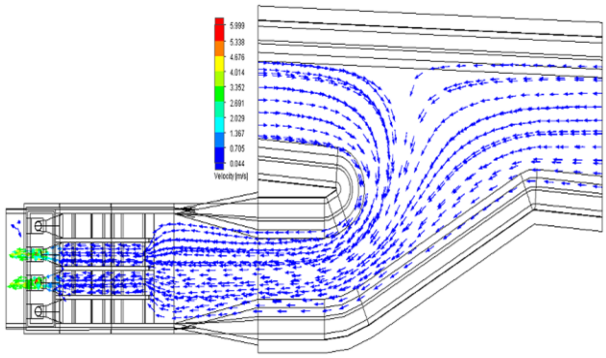
The relative pressure is calculated as the difference between the actual pressure and the reference pressure (atmospheric). If negative, the water would tend to go below the level imposed. Figure (6) presents the relative pressure plots at pump flow rate equal to 6.00 m3/s and suction water level of -3 m for each pump when running pumps number 2, 3 and 4 together. In general there is no negative pressure in pump sump. Also from figure (6) it can be illustrated that, the most critical instants are at the end of activation phase of each pump; the effect remains visible, but not worrying, at the activation of pump 2, while tends to dampen when pump 3 is switched on. The most critical case is the one at pump flow rate equal to 6.00 m3/s and the suction water level of -3 m. However, in all cases, the relative pressure is positive values, and therefore water doesn’t tend to go below the imposed limit.



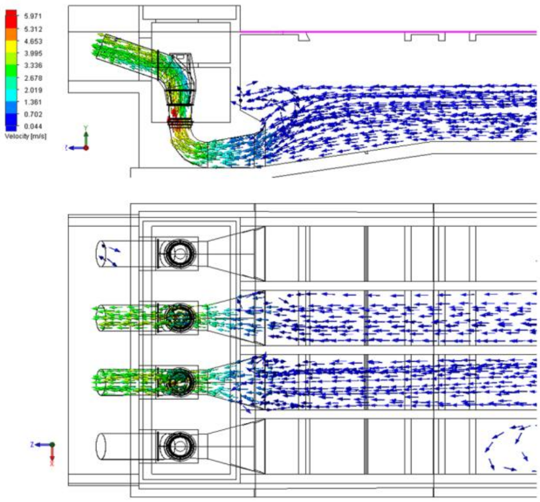
(a) at 20 sec-top view



(b) at 300 sec- top view



(c) at 320 sec



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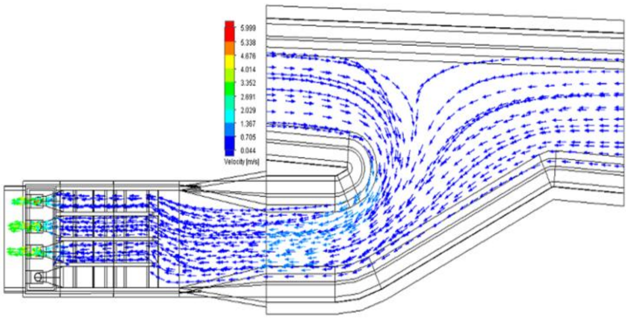

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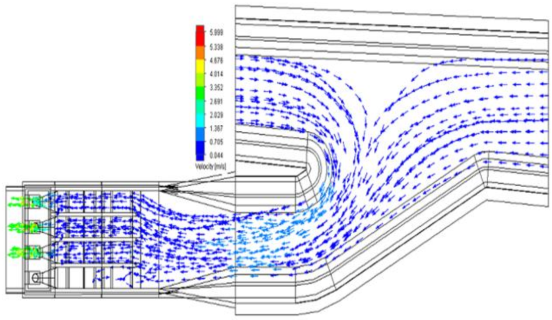
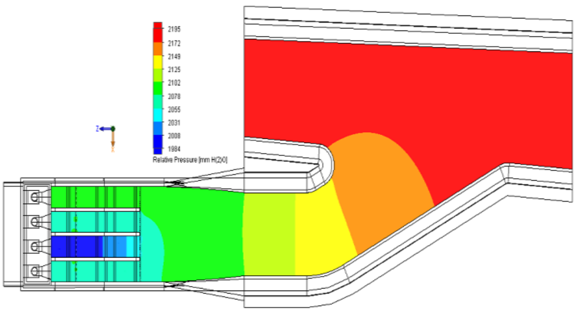
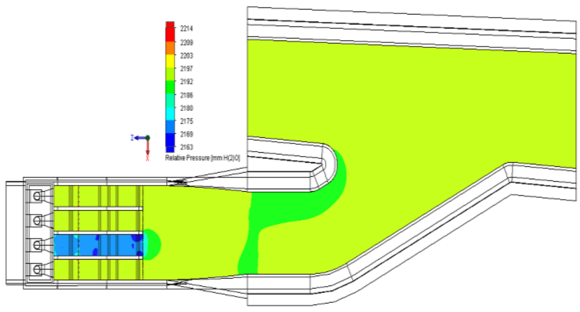


(f) at 900 sec

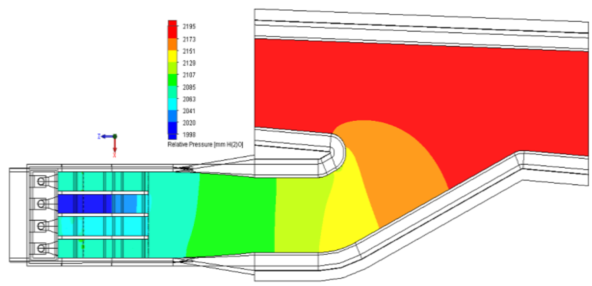
Figure 5: Velocity distribution (m/s) at suction water level of -3 m with different time periods



(a) at 20 sec



(b) at 300 sec



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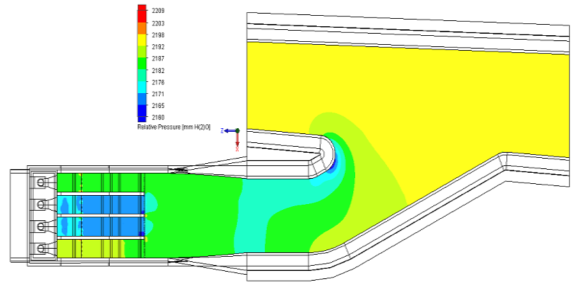

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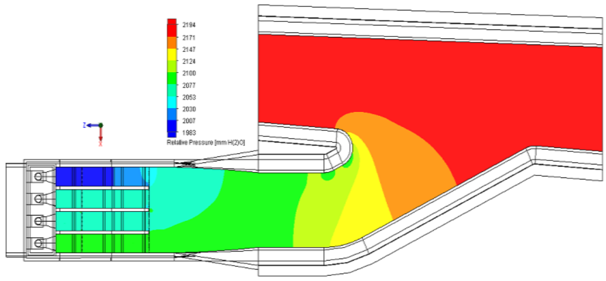

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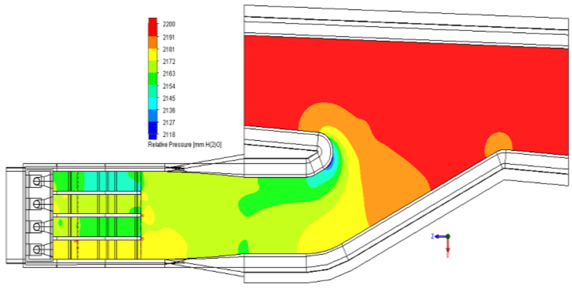


(f) at 900 sec

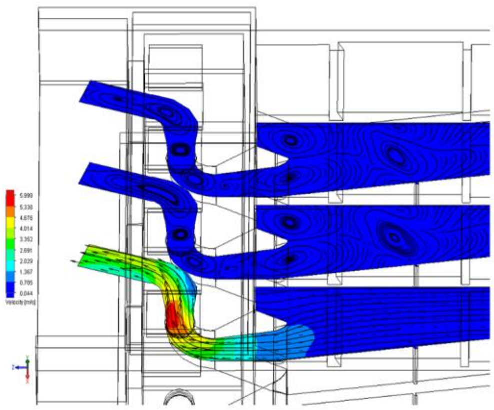
Figure 6: Relative pressure distribution (mm $H_2o$) at suction water level of -3 m with different time periods

**5.2.3 Contour of velocity plots at pump flow rate equal to 6.00 m$^3$/s &suction water level of -3 m**

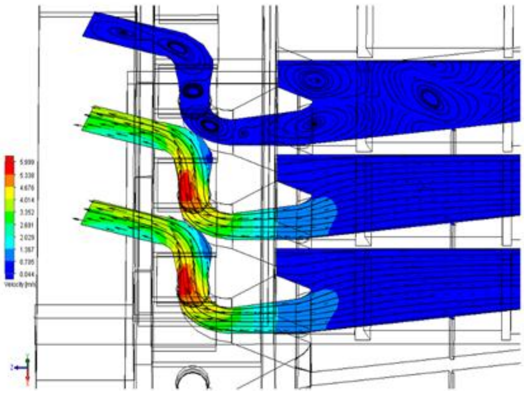
Figure (7) illustrates the contour of velocity magnitude at different time step. Figure (7-a) presents the velocity magnitude when running unit No. 2. From this figure in can notice that, the maximum velocity reached to 5.99 m/s at the suction pump pipe without any flow vortices. Velocity was very high at the suction pump pipe. Red color indicates that the velocity, could achieve from 5.33 m/s to 5.99 m/s at this area. The result shows that the contour of velocity magnitude was symmetrically at each pump suction pipe. The water flow shows a slight recirculation at 300 s (right before the activation of pump 3) on the duct of pump two and at 600 s (right before the activation of pump 4) on the duct of pump 3. The phenomenon does not involve the mid plane of the pumps but the lateral areas where there is little suction as shown in figure (7-b). The water flow shows a slight recirculation at 300 s (right before the activation of pump 3) on the duct of pump 2 and at 600 s (right before the activation of pump 4) on the duct of pump 3. The phenomenon does not involve the mid plane of the pumps but the lateral areas where there is little suction as shown in Figure (7-c).

**5.2.4 Velocity distribution at pump flow rate equal to 6.62 m$^3$/s &suction water level of -2.2 m**

Figure (8) shows the velocity distribution at pump flow rate equal to 6.62 m$^3$/s when suction water level reached to -2.2 m. The maximum velocity is decreased from 5.99 m/s to 5.44 m/s that means the approached velocity is decrease compare to the first scenario. Consequently the flow is reached to all running pumps and there is no circulation zones appeared at the pump sump. This means that the sump hydraulic conditions are improved when suction water level in the sump is decreased.



(a) at 300 sec



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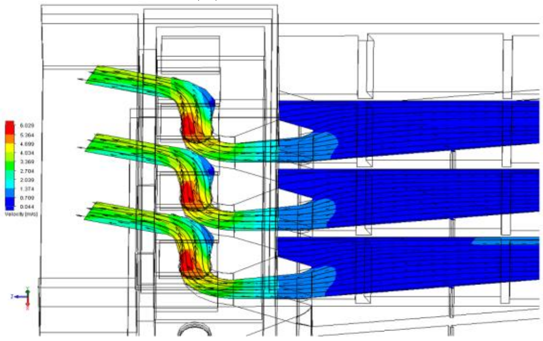
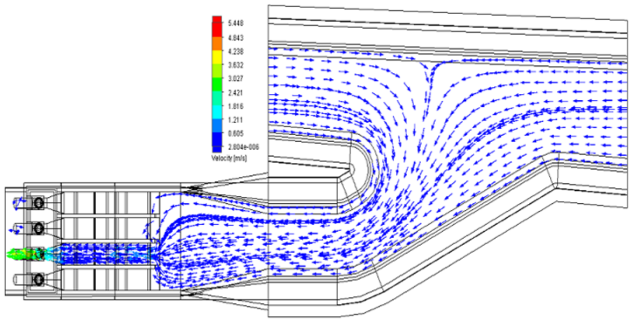


(c) at 900 sec

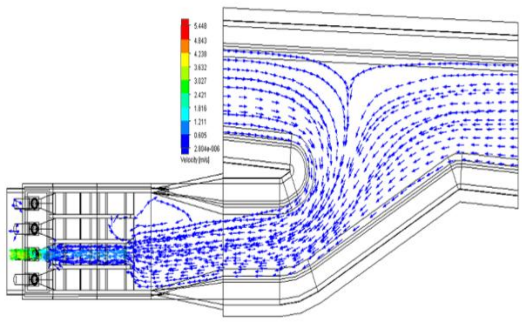
Figure 7: Contour of velocity plots (m/s) at suction water level of -3 m with different time periods

**5.2.5 Relative pressure plots distribution at pump flow rate equal to 6.62 $m^3/s$ &suction water level of-2.2 m**

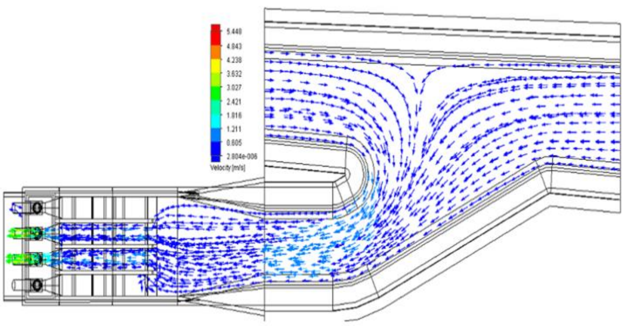
Figure (9) shows the relative pressure plots at pump flow rate equal to 6.62 $m^3/s$ and suction water level, WL=-2.2 m for each pump when running pumps number 2,3 and 4 together. Also there is no negative pressure in pump sump similarly as the first scenario with pump flow rate reached to 6.00 $m^3/s$. Also from Figure (9) it can be illustrated that, the relative pressure in all sump pump increases compared with first scenario. On the other hand side the pressure at the inlet for each running pump also increase. That due to increase the sump suction water level from -3 m to -2.2 m. That leads to when decreasing the suction water level the relative pressure and all hydraulic parameters is improved.



(a) at 20 sec-top view



(b) at 300 sec- top view



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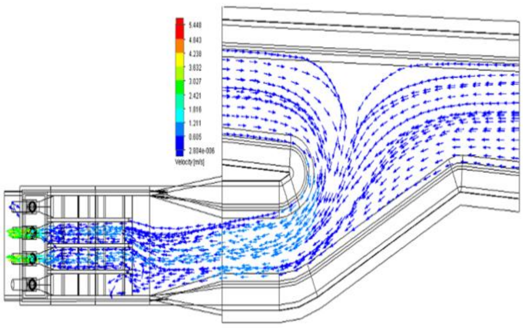

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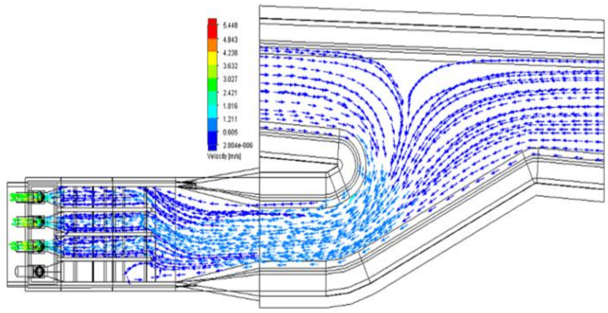
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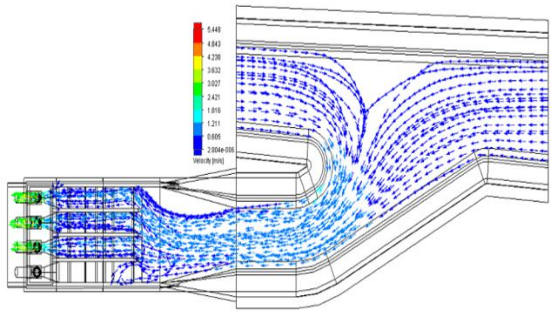
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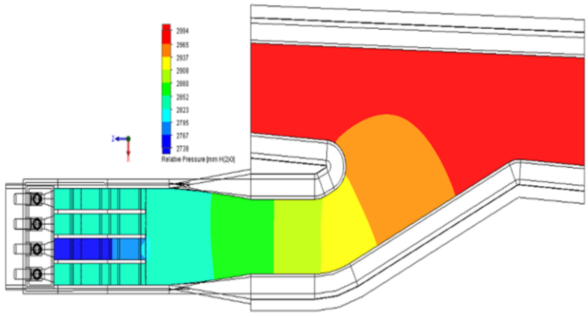
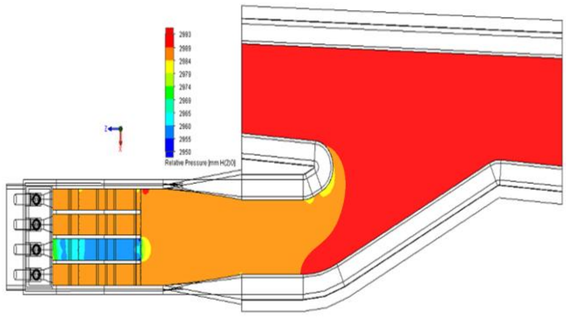
Figure 8: Velocity distribution (m/s) at suction water level of -2.2 m with different time periods


(a) at 20 sec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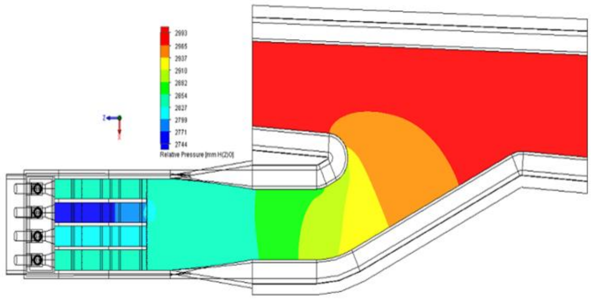
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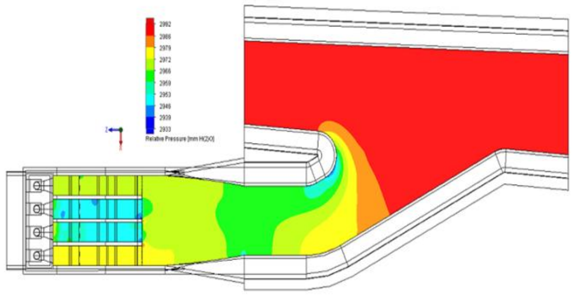
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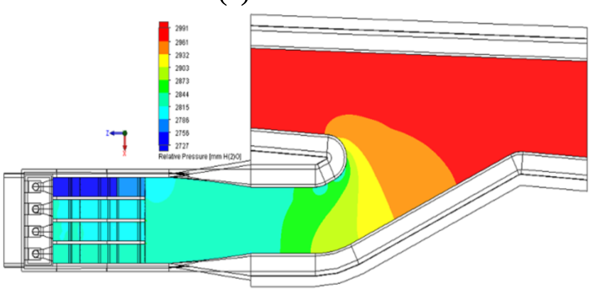
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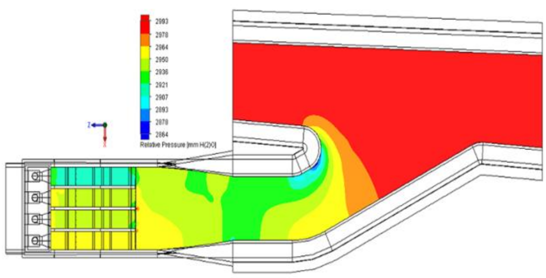
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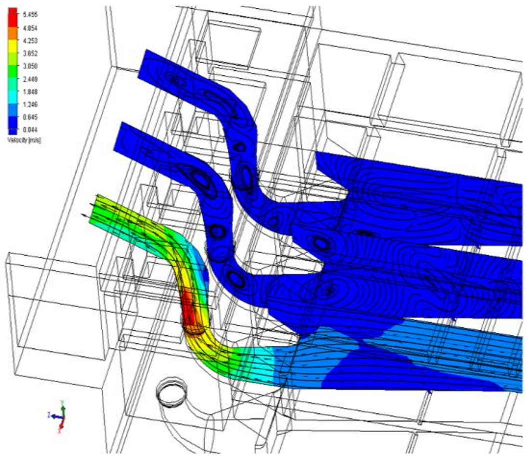
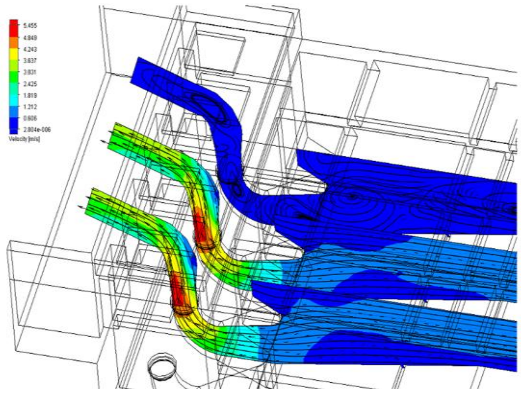
Figure 9: Relative pressure distribution (mm $H_2o$) at suction water level of -2.2 m with different time periods

### 5.2.6 Contour of velocity plots at pump flow rate equal to 6.62 $m^3/s$ &suction water level of-2.2m

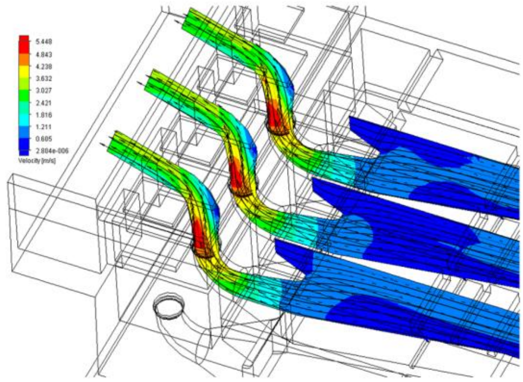
Figure (10) shows the contour of velocity magnitude at different time step when pump flow rate equal to 6.62 $m^3/s$ at suction water level, WL=-2.2 m. From Figure (10) it can be illustrated that the maximum velocity decreased from to 5.99 m/s to 5.44 m/s compared with the first scenario. From this figure it can notice that, the maximum velocity reached to 5.44 m/s at the suction pump pipe without any flow vortices. Velocity was very high at the suction pump pipe. Red color indicates that the velocity, could achieve from 4.85 m/s to 5.45 m/s at this area. CFD software is a good hand tool to investigate a velocity distribution through the pump intake, evaluate suction sump geometry and simulate hydraulic problems. The CFD could be used to simulate flow characteristics in pump sump as an alternative to physical model. From simulation results the geometry of suction intake is proper for running three parallel axial flow pumps with the designed flow rate. On the other hand, velocity counter shows a slight recirculation at 300 s, (right before the activation of pump 3), on the duct of unit No. 2 and at 600 s, (right before the activation of pump 4) on the duct of unit No. 3. So the vibration measurements and analysis should be done.



(a) at 300 sec



(b) at 600 sec



(c) at 900 sec

Figure 10: Contour of velocity plots (m/s) at suction water level of-2.2 m with different time periods

### 5.2.2 Vibration Measurements

Vibration due to mechanical problems can be controlled through more precise balancing of the rotating elements. Vertical pumps can exhibit high vibration levels than horizontal mounted pumps. These pumps often operate with signs of unstable operation, large misalignments, and other characteristics that would cause immediate shutdown in most machinery, [15, 16, 17 and 18]. Unbalanced rotors are a very common cause of machinery malfunction. An improperly balanced machine has many hidden costs in downtime and parts due to accelerated wear and performance issues, [19 and 20]. Forced vibration analysis was done to specify the sources of vibration and define the exciting operational frequencies at different locations. The measurement location were done on the parts of the pumping unit including the motor, bearings, and the pump at 12 locations in the axial, vertical, and radial directions as shown in Figure (11). The measurements locations and directions are:

-Point (1, 2 and 3) motor non drive end in horizontal, vertical and axial direction (MNDEH, V and A)
-Point (4, 5 and 6) motor drive end in horizontal vertical and axial direction (MDEH, V and A).
-Point (7, 8 and 9) gearbox in horizontal, vertical and axial direction (GDEH, V and A).
-Point (10, 11 and 12) pump drive end in horizontal, vertical and axial direction (PDEH, V and A).

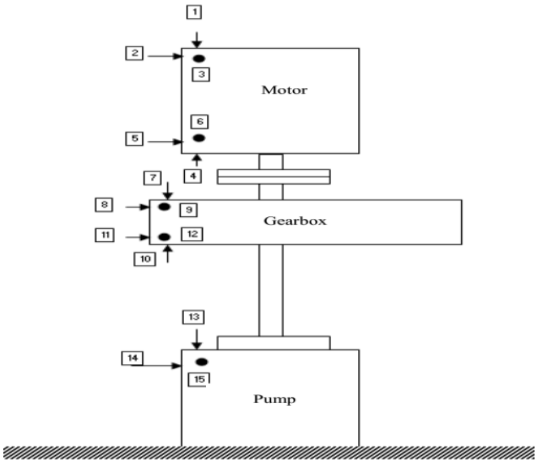



Figure 11: Location of measurement points

### 5.2.3 Dynamic Stability Results

Vibration measurement provides sound bases for establishing the running condition of the pumps. All these result were taken following the ISO 10816-3 [21] in the two cases of different discharges and after balancing. The motor non drive end overall vibration levels are within the acceptable range according to ISO 10816-3 at flow rate 6.62 $m^3/s$ and it reached to 1.23 mm/s. The measurements on the motor non drive end at flow rate 6.00 $m^3/s$, showed that the maximum overall vibration level measured reached 3.88 mm/s at the motor non drive. According to ISO 10816-3 [21] the vibration levels increased by 215 % in vertical direction end and it slightly dangerous values are listed in Table (1). On the other hand the measurements on the pump drive end showed that the maximum overall vibration level measured at flow rate 6.62 $m^3/s$ reached to 0.71 mm/s.

Table (1) Overall vibration level velocity measurements for unit No.2

| Cases | | Q= 6.00 $m^3/s$ and WL= -3 m | | Q= 6.62 $m^3/s$ and WL= -2.2 m | |
|---|---|---|---|---|---|
| Measurement locations | | Overall vibration velocity, (mm/s) before balancing | Overall vibration velocity, (mm/s) after balancing | Overall vibration velocity, (mm/s) before balancing | Overall vibration velocity, (mm/s) after balancing |
| Motor | MNDEH | 2.35 | 1.2 | 0.97 | 0.63 |
| | MNDEV | 3.88 | 0.95 | 1.23 | 0.65 |
| | MNDEA | 0.47 | 0.43 | 0.53 | 0.50 |
| | MDEH | 1.76 | 0.9 | 0.95 | 0.61 |
| | MDEV | 4.01 | 1.01 | 1.35 | 0.9 |
| | MDEA | 0.66 | 0.62 | 0.55 | 0.51 |
| Gearbox | GDEH | 1.28 | 1.15 | 0.74 | 0.72 |
| | GDEV | 1.33 | 1.18 | 0.84 | 0.80 |
| | GDEA | 1.88 | 1.52 | 0.62 | 0.59 |
| Pump | PDEH | 1.13 | 1.05 | 0.71 | 0.68 |
| | PDEV | 1.40 | 0.9 | 0.47 | 0.40 |
| | PDEA | 1.60 | 1.2 | 0.60 | 0.58 |

The motor drive end overall vibration levels are within the acceptable range according to ISO 10816-3. The measurements on the motor drive end at flow rate 6.00 m³/s showed that the maximum overall vibration level measured reached 4.01 mm/s, on the motor drive end. According to ISO 10816-3, the vibration level was increased by 197 % in vertical direction. The Fast Fourier Transform (FFT) spectrum analysis have been recorded that, there are vibration peaks founded at motor running speed (1RPM, 25Hz) in the horizontal and vertical directions of the motor. Vibration amplitude reached to 3.88 mm/s, and 4.01 mm/s at the motor non drive end and the vertical direction at motor drive end respectively at flow rate 6.00 m³/s are presented in Figures (12 and 13). These high levels of vibration are in the danger zone according to ISO 10816-3 [21]. The results indicated that the amplitude changed with change the suction water level at the sump and flow rate. Depending on all these remarks, it is obvious that there is an unbalance problem. Balancing has done for each unit and presented results here are for unit No. 2. The vibration level decreased to 1.01 mm/s from 4.01 mm/s at flow rate equal to 6.00 $m^3$/s. On the other hand the vibration level decreased to 0.9 mm/s from 1.35 mm/s at flow rate equal to 6.62 $m^3$/s, when added weight equal to 200 gm at motor fan. Unbalance problem causes an overload on bearings and results in shortening bearing life [1]. This reduction depends on the degree of unbalance. Furthermore the result of the coupling bore not being perfectly centered or being off angularly. If the coupling mass is located only a few mils off-center, it creates static unbalance. If the bore angle is not square with the coupling faces within a few mils, it causes couple unbalance. Most often the coupling will increase the total assembly's unbalance. Misalignment produces excessive vibration, noise, coupling, and bearing temperature increases, and premature bearing, coupling, or shaft failure.

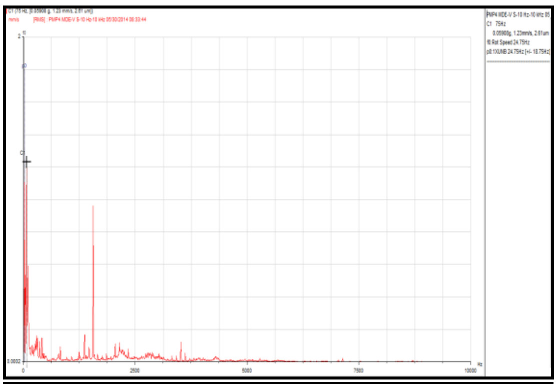


Figure 12: Vibration spectrum measured on motor in vertical direction at flow rate 6.62 $m^3$/s

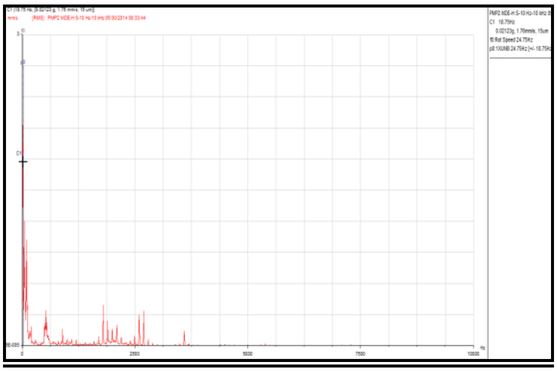


Figure 13: Vibration spectrum measured on motor in horizontal direction at flow rate 6.00 $m^3$/s

## VI. CONCLUSION AND RECOMMENDATIONS

Experimental study and Numerical simulations are performed to investigate dynamic stability of Zaghloul pumping stations. Calculations are performed when running three pumping units in parallel. The following statements summarize more important conclusions:

1. The geometry of suction intake is proper for running three parallel axial flow pumps with the designed flow rate. Activation more than two units lead to decrease the suction water level, disturbance of velocity distribution in suction intake and vibration severity level increased.

2. The velocity distribution in the case of pump flow rate equal to 6.00 $m^3$/s at suction water level equal to -3 m shows a slight recirculation at 300 s (right before the activation of pump three) on the duct of pump two and at 600 s (right before the activation of pump four) on the duct of pump three. The phenomenon does not involve the mid plane of the pumps but the lateral areas where there is little suction.

3. The maximum velocity is decreased from 5.99 m/s to 5.44 m/s at suction water level equal to -2.2 m, which means the approached velocity, are decrease compare to the suction water level equal to -3 m. Consequently the flow reached to all running pumps and there is no circulation zones appeared at the pump sump. This means that the sump hydraulic conditions are improved when suction water level in the sump decreased.

4. In all studied cases; the relative pressure values are positive, the most critical case is the one at pump flow rate equal to 6.00

$m^3/s$ at the suction water level of -3 m. Also the most critical instants is at the end of activation phase of each pump; the effect remains visible, but not worrying, at the activation of pump two, while tends to dampen when pump three is switched on.

5. The balancing reduce vibration level and save bearing life, the vibration level decreased from 4.01 mm/s to 1.01 mm/s when suction water level reached to -2.2 m at flow rate 6.00 $m^3/s$. On the other hand the vibration level decreased from 1.35 mm/s to 0.95 mm/s when suction water level reached to -3 m at flow rate 6.620 $m^3/s$.

It is recommended that, dynamic analysis and balancing should be done at the primary operation and evaluation for pumping station stability. Also must check the working operation range for (discharge-head) curve.

## REFERENCES

[1] S. M. Abdel-Rahman and Sami A. A. El-Shaikh, "Diagnosis Vibration Problems of Pumping Stations: Case Studies", 13th International Water Technology Conference, Hurghada, Egypt, pp.419-434, 2009.
[2] S. Tanweer Desmukh and M. Gahlot "Simulation of Flow through a Pump Sump and its Validation", IJRRAS 4 (1), pp.7-17, July 2010.
[3] Mohd. Remy Rozainy and M. I. Mohd. Ashraf, "Application of Computational Fluid Dynamics (CFD) in Physical Model of Pump Sump to Predict the Flow Characteristics", ICCBT08, D - (07) – pp.79-90, 2008.
[4] S.M. Borghei and A.R. Kabiri-Samani, "Effect of Anti-Vortex Plates on Critical Submergence at a Vertical Intake", Transaction A: Civil Engineering, Vol. 17, No. 2, pp. 89-95, Sharif University of Technology, April 2010.
[5] Y X Zhao1, C G Kim1 and Y H Lee, "CFD Study on Flow Characteristics of Pump Sump and Performance Analysis of the Mixed Flow Pump", 6th International Journal of Pumps and Fans with Compressors and Wind Turbines, 2013.
[6] V. P. Rajendran, S. G. Constantinescu and V. C. Patel, "Experiments on Flow in Model Water pump Intake Sump to Validate a Numerical Model", Proc. of ASME Fluids Engineering Division Summer Meeting FEDSM (Washington, USA, 21–25June 1998), pp. 21-25 ,1998.
[7] R. Iwano, T. Shibata, T. Nagahara and T. Okamura, "Numerical Prediction Method of a Submerged Vortex and Its Application to the Flow in Pump Sumps with and without a Baffle Plate", Proc. of the 9th Int. Symp. on Transport Phenomena and Dynamics of Rotating Machinery (Honolulu, Hawaii, USA, 10-14 February 2002)1-6 ,2002.
[8] J. W. Choi, Y. D. Choi, C. G. Kim and Y. H. Lee, " Flow Uniformity in a Multi Intake Pump Sump Model", Journal of Mechanical Science and Technology 24 (7) 1389-1400, 2010.
[9] Y. H. Lee, "Establishment of Design Guideline for the Pump Intake Shape using the Result of Model Test", R&D Report, K-water (in Korean), 2004.
[10] Turbomachinery Society of Japan, “Standard Method for Model Testing the Performance of a Pump Sump”, TSJ S002, 2005.
[11] C. G. Kim, J. W. Choi, Y.D. Choi and Y. H. Lee, "A Study on the Effectiveness of an Anti-Vortex Device in the Sump Model by Experiment and CFD", 26th IAHR Symposium on Hydraulic Machinery and Systems, 2012.
[12] Joseph Orlins, Zachary Taylor and Justn Arnold, “Comparison of Physical Hydraulic and CFD Model Measurements of Velocity and Swirl Angle In Pump Intakes”, E-proceedings of the 36th IAHR World Congress, The Hague, the Netherlands, 28 June – 3 July, 2015.
[13] B. E. Launder and D. B. Spalding, "The Numerical Computation of Turbulent Flows", Computer Methods in Applied Mechanics and Engineering, Vol. 3, pp. 269-289, 1974.
[14] Chis Timur, “Modern Pipe Line Monitoring Technologies”, Fist International Symposium of Flow Measurement and Control, Tokyo, Japan, 2005.
[15] R. Smith and G. Woodward, “Vibration Analysis of vertical pumps, "Sound and Vibration, Vol. 22, No. 6, pp. 24-30, 1988.
[16] T. Walter, M. Marchonie and H. Shugars, "Diagnosis Vibration Problems in Vertically Mounted Pumps", Transactions of the ASME, Vol. 110, pp. 172-177, April, 1988.
[17] M. A. Younes, S. M. Abdel-Rahman and M. A. Helal, “Dynamic Response and Instability of Propeller Pumps", 21th International Conference of the Dynamic Analysis, Vibration and Organized by the Association of Experimental Mechanics

(SEM) in Kizmy - Florida - United States, February 2003.

[18] A.W. Lees, "Fault Diagnosis in Rotating Machinery", 18th International Modal Analysis Conf. (IMAC), San Antonio, Texas, pp. 313-319, Feb 2000.

[19] Eric Li and Suri Ganeriwala, "Vibration and Force Signatures of Overhung Rotor Rotating Machine with Unknown Initial Conditions," Tech. Notes, Spectra Quest Inc., VA 23228, July 2008.

[20] T. Ralph Buscarello, "Practical Solutions to Machinery and Maintenance Vibration Problems", Update International; Revised edition, May 2002.

[21] ISO 10816-3, 1995, "Mechanical Vibration– Evaluation of Machine Vibration by Measurements on Non-Rotating Parts", Part 1, General Guidelines