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EXPERIMENTAL INVESTIGATION AND FINITE ELEMENT ANALYSIS FOR SOLVING VERTICAL PUMPS STRUCTURAL WEAKNESS

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ABSTRACT: *Pump stations are generally exposed to various mechanical and structural problems causing vibration affecting efficiency, performance, reliability life, and maintenance cost. In this research, Vibration problem of Helwan Irrigation Pumping Station is studied and analyzed to define vibration sources leading to damaging the foundation of most pumps in the station. A finite element model of Helwan Irrigation Pumping Station was built. Vibration level and frequency analysis are measured on the machinery parts and on the foundations of the pumping station at different locations to overcome the high level of vibration, solve the resonance problem, determine the source of vibration, and the path of vibration transmission to the foundation. The level of vibration measured 16.4 mm/sec at (MNDE-H). Frequency analysis shows that there is resonance problem. The model suggests different scenarios to solve the structural problem and overcome the resonance. Applying the different scenarios and measurements are repeated until the problem disappeared where the level of vibration is in the range of 3.9 mm/sec (allowable). The Rresults point out that adding supports to weak motor foundation enhance the dynamic characteristics and keeping the pumping stations in smooth running conditions. Inspection and regular maintenance is important to avoid any abnormal conditions.*

KEYWORDS: Pump Vibration, Support Weakness, Dynamic Performance

INTRODUCTION

Vibration is one of the most serious problems in the pumping stations in Egypt. Pumping stations are subjected to many operational problems including mechanical and hydraulically problems affecting the performance and efficiency of the pumping stations. It is very important to avoid or overcome these problems to obtain maximum efficiency for the pumping station. Mechanical sources including unbalance, misalignment, bearing, resonance, and etc... Pumping stations, especially vertical pumps, are sensitive to mechanical and structural problems leading to vibrations. To avoid these high levels of vibrations for vertical pumps, resonance problem must be prevented. Axial flow pumping system is usually used to deliver high discharges at low heads. This pumping system is currently used on large scale for reclamation purposes of new lands in Egypt. Vertical and inclined installations of these pumping systems introduce vibration problems of such plants. Axial Flow Pumping stations always have long rotating shafts between the motor, the gearbox, and the pump working as a cantilever fixed at the bottom (pump) and free at the top (motor). So, when analyzing vibration in a pumping system, it is necessary to look at all components of the system and determine which component may cause the problem to the whole system [1].

The purpose of modeling pumping units is to show its internal workings and to present it in a form useful to engineering study. Modeling of pumping units can be done using scale models

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for laboratory testing, and mathematical models including computer, analytical, linear, and nonlinear models. Modeling of the pumping units can be done using lumped-mass models and finite element analysis to predict dynamic behavior at different conditions and to obtain dynamic characteristics of the system. Experimental modal testing using finite element method and lumped mass model is done to determine eigenvalues and eigenvectors representing dynamic characteristics of the pumping system [2].

Jyoti [3] conducted Modal analysis on the complete assembly of pumps, piping layout and identified resonance as the root cause for pump failure [3]. Tony [4] stated that modal techniques are powerful tools that enhance an analyst's ability to understand the sources of vibration. A case history of vertical pump was investigated. Testing progression from problem identification in route vibration measurements to resonance testing was presented. Resonance problems are difficult to solve. . Modal Analysis give a clear picture of the machine's motion, however neither tool has the capability to solve resonance problems [4].

Cornelius [5] conducted an experiment to monitor pump condition through vibration analysis. This research illustrates the typical steps required to solve resonance problems. The paper describes the use of operational deflection shape (ODS) and modal analysis testing for problem solving [5

There has been considerable interest in the maintenance techniques based on condition monitoring, with the analysis of vibration characteristics generated by machines, which makes it possible to determine whether the machinery is in good or bad condition[6].

In recent years, Condition Monitoring is defined as the collection, comparison and storage of measurements defining machine condition. Almost everyone will recognize the existence of a machine problem sooner or later. One of the objectives of Condition Monitoring is to recognize damage that has occurred so that ample time is available to schedule repairs with minimum disruption to operation and production [7].

High amplitude vibration levels can cause damage to the building structures and components. When vibration is destructive to building component the vibration will be highly perceptible to the building occupants. Most available guidelines are based on frequency-velocity control limits. Studies have shown that velocity seems to correlate closely with observed damage. Frequency plays a large role in vibration related structural damage. Common structures have a low natural frequency, typically less than 30 Hz. Structural vibration is exponentially increased if the vibration frequency falls within the bounds of the natural frequency of the structure. This phenomenon is commonly known as resonance. Thus, low frequency vibrations are potentially more of a concern than their high frequency counterparts. A vibration velocity of 25 mm/sec is used as a normally safe vibration upper limit with respect to structural damage. Vibrations with a velocity level greater than 25 mm/sec should be avoided or special arrangements should be made. Even with a vibration level of 25 mm/sec superficial damage may occur in isolated instances. In order to ensure that the possibility of superficial damage is minimized a vibration criteria of 5 mm/sec has been recommended. For very old structures, an even lower level of 1.25 mm/s is recommended [8]. A 0.07 g (acceleration of gravity) vibration is recommended as a safe limit of large structures [10].

Doebling[11] present a comprehensive literature review of damage identification and health monitoring methods for structural and mechanical systems focusing on methods based on

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vibration measurements and detection based on changes in vibration characteristics. Rytter [12] defines four stages of damage monitoring:

(1) Determination that damage is present in the structure;
(2) Determination of the geometric location of the damage;
(3) Quantification of the severity of the damage; and
(4) Prediction of the remaining service life of the structure.

PROBLEM STATEMENT AND METHODOLOGY

The vibration problem of Helwan irrigation pumping station damages the structure foundation for most pumps. In this research, a resonance problem of Helwan pumping station was assessed. Helwan Pumping Station is used to serve irrigation of 20000 feddans in Teraat El-Saf in Helwan City, consisting of 10 pump units as shown in **Figure (1)**. Each pump unit is of discharge 2 m^3/sec, head 5 m, pump speed 500 rpm, motor power 900 kW, motor speed 744 rpm, motor weight 8500 kg, gearbox ratio 2.48, number of drive Gear teeth is 25 Pinion Gear, number of driven Gear teeth is 62, and rated power for gearbox 456 KW. Equipment that used for test/measurement is *one proD/ACOEM* vibration analyzer and *Data collector MVP200* serial 11141 with Machine Monitoring SW type *XPR300 Premium*, as shown in **Figure (2)**.

Figure (1) "Photograph for Helwan P. S. "

Figure (2) "01dB Movipack with machine monitoring SW type XPR300 premium"

RESEARCH METHODOLOGY

In this research, theoretical model for pumping station by using Modal Shape program was done to define the places high of stresses on the various parts of the motor and foundation to overcome the structural weakness that leads to resonance problems. Vibration measurements were taken to determine the status of the system, and also Solid Work program was used to draw this pumping station to represent the problem in theory.

Theoretical model analysis program was used to determine natural frequencies for the system and to assist the pump to operate away from these natural frequencies for the system. Frequency analysis is used to determine the sources of high vibration that affect the operation of the pumping station, by knowing dynamic characteristics (which include: natural

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frequencies, damping, and mode shape) of the pumping system. The model was applied on an axial flow pumping unit to control of dynamic behavior of the pumping unit by introducing structural modifications. Optimum structural changes and sensitive nodes are well defined for modifying the system dynamic behavior. A mathematical model of the pumping unit is defined and solved to obtain the dynamic characteristics of the pumping system. Good correlations between the experimental and analytical modal analyses are obtained.

Dynamic Analysis is done to indicate the severity of vibration which is the basis to judge the status of machinery if the conditions within the standard or not. Frequency Analysis is done to know high unwanted frequencies for each element in the pump system, to define the exciting frequencies and determine the level of vibration at each specific frequency, to determine the sources of vibration, to control vibration levels, and to solve vibration problems. Firstly, Dynamic Analysis by measuring overall vibration velocity, vibration acceleration, and bearing defect factor were done at 9 locations on all units in three directions axial, horizontal, and vertical perpendicular, as shown in **Figure (3)** and compared the results with standards of machines (ISO 10816-3) **[13]**, as shown in **Figure (4)**. Frequency analysis is done at low and high frequency. The second step was applying the several scenarios from the theoretical model program to solving these vibration problems. From applying the results of first scenario (first modification) which suggests to making modification for the concrete foundation by adding steel foundation with depth 240 mm, width 240 mm, web thickness 10 mm, flange thickness 17 mm, and fillet radius 21 mm, it was found that the overall vibration level decrease from 16.5 mm/sec to 8.9 mm/sec with about 40% danger than before. By applying the results of second scenario (second modification) which suggests to increase modification for the steel foundation with depth 320 mm, width 300 mm, web thickness 11.5 mm, flange thickness 20.5 mm, and fillet radius 27 mm, it was found that the overall vibration level decrease with about 80% than before where it decrease from 8.9 mm/sec to 3.9 mm/sec,as shown in **Figure (5)**.

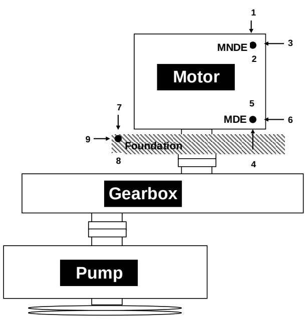

Figure (3) "Measurement locations"

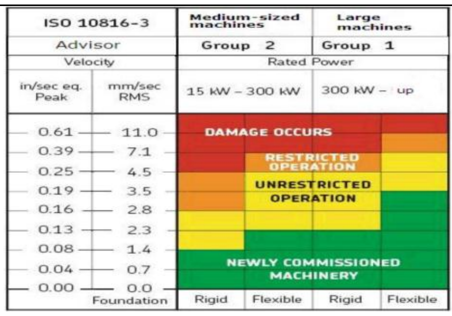

Figure (4) "ISO Standard 10816-3"

Bearing Defect Factor

Bearing Defect Factor coefficient also was measured to judge the status of bearing cases and knows the defect bearing, as shown in **"Table [1]"**.

Figure (5-a) "Foundation before modification"

Figure (5-b) "Foundation after modification"

Figure (5) "Foundation before and after modification"

Machine condition	Velocity (mm/sec)	Acceleration (g)	Bearing Defect Factor (BDF)
Acceptable	**Level < 4.5 mm/sec**	**Level < 1 g**	**Level < 6**
Alarm	**4.5 < level < 7.1 mm/sec**	**1 < level < 1.5 g**	**6 < level < 9**
Danger	**Level > 7.1 mm/sec**	**Level > 1.5 g**	**level > 9**

Table [1] "Bearing Defect Factor coefficient"

RESULTS & ANALYSIS

Forced vibration analyses were done by measuring vibration in three directions axial, radial, and radial perpendicular as shown in **Figure (3)** on all units of Helwan Irrigation pumping station to specify the sources of vibration and to define the exciting operational frequencies. All measurements were taken with root mean square velocity (RMS) in a frequency range from 2 Hz to 1 kHz and the results compared with the ISO 10816-3, as shown in **Figure (4)**, where it is the most accommodate one for machine diagnosing as it expresses for the energy consumption due to vibration also it takes the time history of vibrations. Where, simple spectrum vibration velocity Fast Fourier Transform (FFT) in a frequency range from 2 Hz to 1 kHz, simple spectrum vibration acceleration was taken from 10 Hz to 10 kHz. The measurement locations are determined on the parts of the pumping station at 9 locations as shown in **Figure (3)**; the overall vibration level measurements for one unit are listed in **Table [2]**.

Dynamic Analysis (before modification)

Overall vibration level was measured and the results were analyzed. Results of measurements show that the vibration level is higher than the ISO standards (ISO10816-3)

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[13], where the maximum level of vibration reached to 16.4 mm/sec on the motor, as shown in **“Table [2]”**.

The measurements results indicated that there are severe high vibration levels on the motor and it’s higher than the standards [11], where the level of vibration is reached to 16.4 mm/sec. Results of one pump unit is only presented and compared with ISO 10816-3, as shown in **Table [1]**. These levels are dangerous and not permissible according to ISO Standard 10816-3 which recommends that for machines with power greater than 300 kW level of vibration, RMS velocity level is good until 2.3 mm/sec; allowable level is until 4.5 mm/sec, and restricted operation level up to 7.1 mm/sec.

Location Measurements	Overall Acceleration (g)			Overall velocity (mm/sec)			Bearing defect Factor (BDF)		
	before Modification	after Mode Modification	after Mode Modification	before Mode Shape	after Mode Shape	after Mode Modification	before Mode Shape	after Mode Shape	after Mode Modification
MNDEH	**1.01**	0.890638	0.538916	**16.408**	**8.9**	**3.9189**	3.9321	3.14571	2.61667
MNDEV	0.612812	0.551531	0.474624	4.2	3.38151	1.8313	4.2896	3.4317	2.76369
MNDEA	0.190237	0.171213	0.086759	2.6168	1.74259	1.5701	3.1322	2.50576	2.22463
MDEH	0.20097	0.180873	0.085932	**4.4280**	**2.87275**	**1.7237**	3.8668	3.0934	2.58398
MDEV	0.276939	0.249245	0.134899	2.1059	1.26359	0.7410	3.4085	2.72682	2.47268
MDEA	0.43571	0.392139	0.29497	2.7391	1.97739	1.6435	3.7203	2.97622	2.31584
Foundation H	**1.1**	0.802271	0.313902	4.9	3.01446	1.9434	5.2079	4.16631	3.90103
Foundation V	0.855003	0.769503	0.422643	**3.2038**	**1.92228**	**0.9286**	5.9538	4.76307	3.75914
Foundation A	0.390191	0.351172	0.187069	2.4857	1.50388	1.4914	5.5249	4.41994	3.16329

Table [2] “Overall vibration levels “

Frequency Analysis (before modification)

Frequency Analysis was done to determine high frequencies for each element in the pump system, and to determine problem at any frequencies. Frequency spectrum analysis indicated that there are vibration peaks at motor running speed 1x (12.4Hz) and its harmonics in the vertical and horizontal directions. It is obviously noticed that the maximum vibration amplitude reached 13.5 mm/sec at 12.5 Hz as shown in **Figure (6-a)**, its harmonics as shown in **Figure (6-b)**, and peaks with high amplitude at low frequency as shown in **Figure (6-c)**. These high levels of vibration are in the not permissible zone of ISO 10816-3 indicating a resonance problem. It is not safe and dangerous to operate at this condition. The cause of this problem is due to the foundation weakness.

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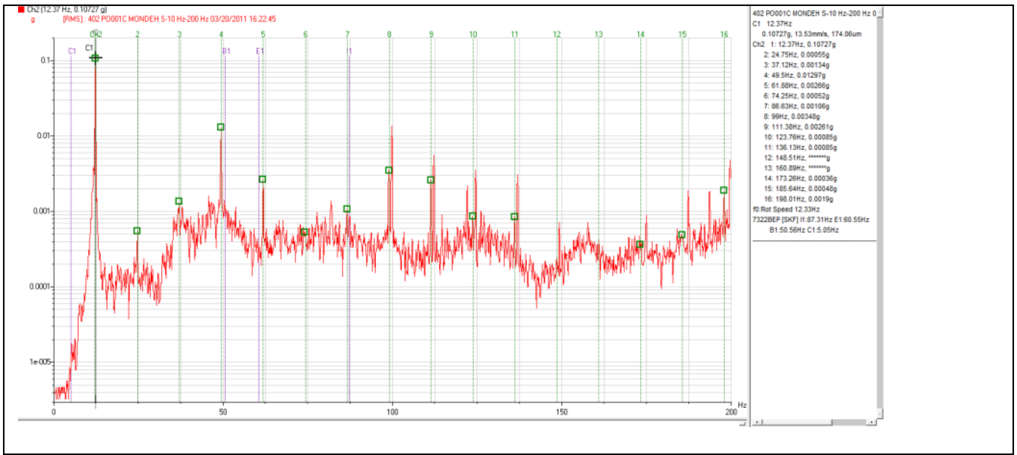

Figure (6-a) “FFT Domain for MNDEH“

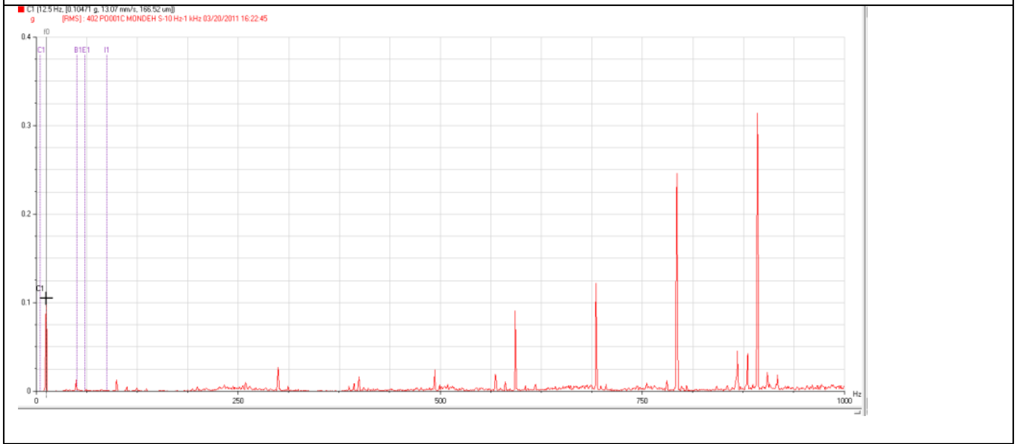

Figure (6-b) “FFT Harmonic for Resonance at MNDEH“

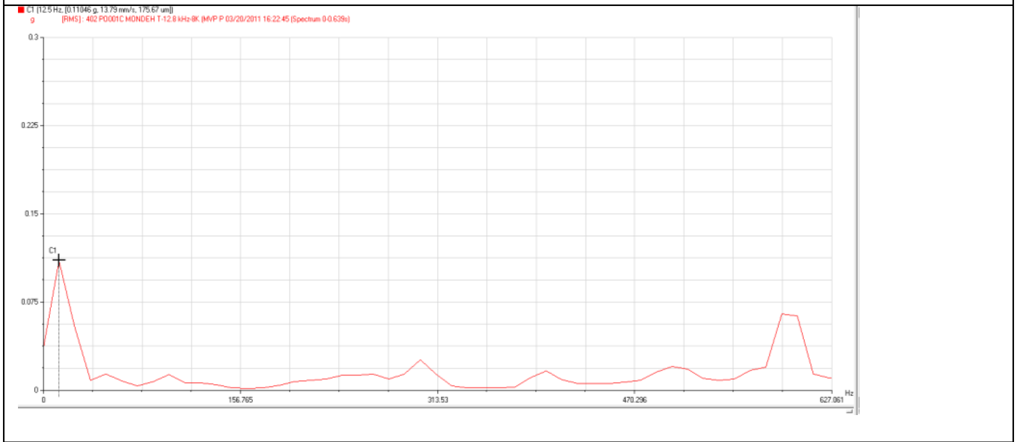

Figure (6-c) “Time domain for Peak at low frequency at MNDEH“

Figure (6) “Frequency Analysis before modification“

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FINITE ELEMENT ANALYSIS

Finite element analysis (FEA) was used to model the motor structure to estimate the dynamic characteristics using ANSYS work bench 14.5. The FEA model was built for the original motor structure and simulation is done to find its natural frequencies and mode shapes.

The model consists of a motor weighing 8700 kg, a steel base, and a concrete foundation. The motor is assumed to be rigid over the frequency range of interest. Concrete is assumed to be a homogeneous and isotropic material and to behave in a linear elastic manner. The mechanical properties of the concrete are assumed to be (Modulus of elasticity: 32 GPa, Poisson’s Ratio: 0.2, and Weight density: 2400 kg/m3). Steel is assumed to be a homogeneous and isotropic material behaving in a linear elastic manner. The mechanical properties are assumed to be (Modulus of elasticity: 200 GPa, Poisson’s Ratio: 0.3, Weight density: 7800 kg/m3). A simple geometrical structure was selected and used to simulate the unconstrained condition using solid elements. When a 3D model of solid volumes is generated, solid modeling is generally more convenient compared to direct generation. Solid modeling is tedious and too much time consuming. A solid model in ANSYS is built from the bottom up by creating key points. They represent the model vertices. They are used to define higher order entities such as lines, areas, and volumes. The boundary conditions assumed that the concrete foundation is fully clamped, as shown in **Figure (7a, 7b, 7c, 7d, 7e, and 7f)**.

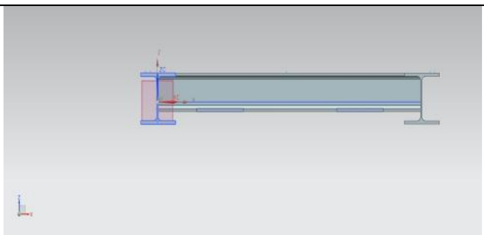

Figure (7-a) **“**Foundation front“

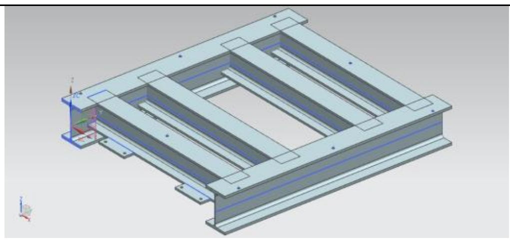

Figure (7-b) **“**Foundation isometric**“**

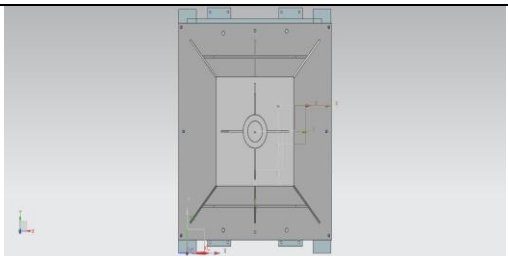

Figure (7-c) “Foundation and pump top“

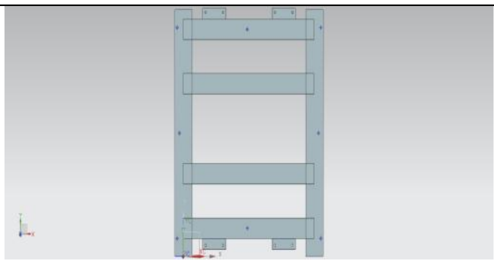

Figure (7-d) “Foundation top“

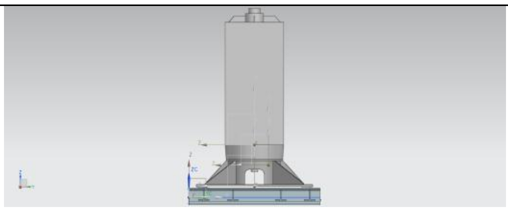

Figure (7-e) “Foundation with motor and pump side“

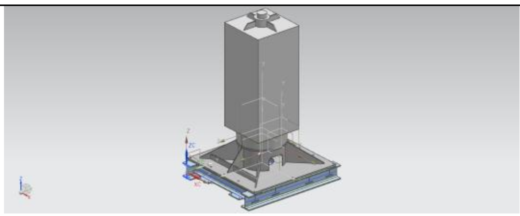

Figure (7-f) “Foundation with motor and pump isometric**“**

Figure (7) “Foundation modification "showing the required base drawing to be fabricated"

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MODEL RESULTS

A simulation was done by the theoretical program to solve the problem that leads to decreasing the level of vibration, enhancing the characteristics of operating pumping station condition. The theoretical program indicates to add Stainless steel supports of motor foundation. Stainless steel support was manufacturer by a certain thickness and height and then, vibrations measurements was repeated again and doing analysis for the results. Modal Shape for motor, and motor foundation before adding support, are shown in **Figures (8a, 8b, 8c, 8d, 8e, and 8f).**

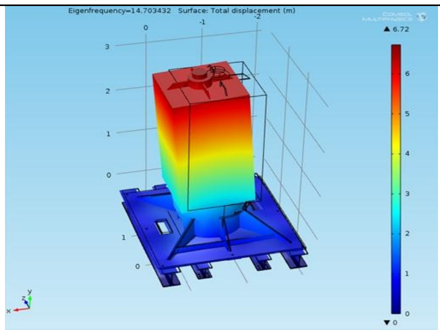

Figure **(a)**

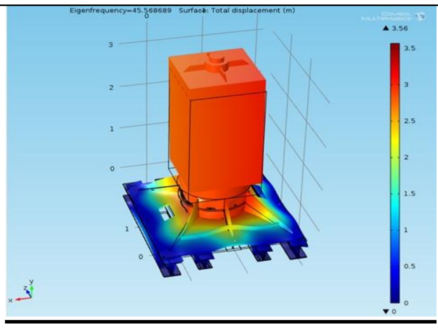

Figure **(b)**

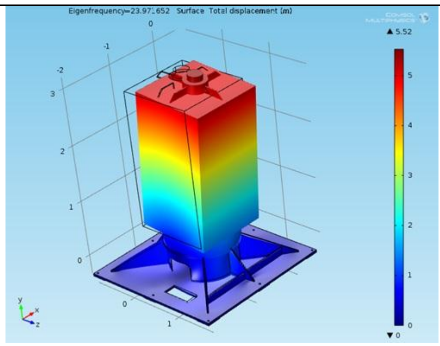

Figure **(c)**

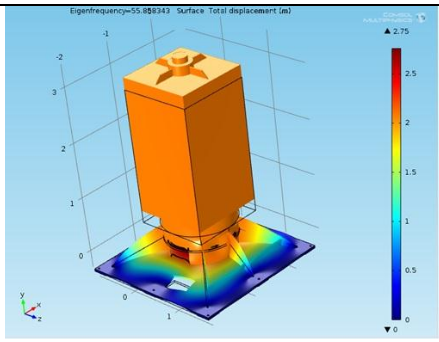

Figure **(d)**

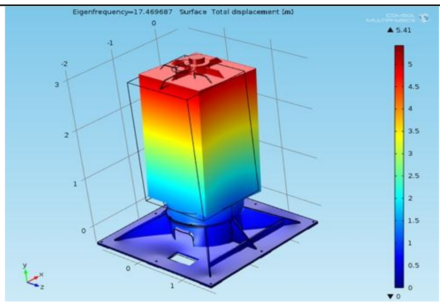

Figure **(e)**

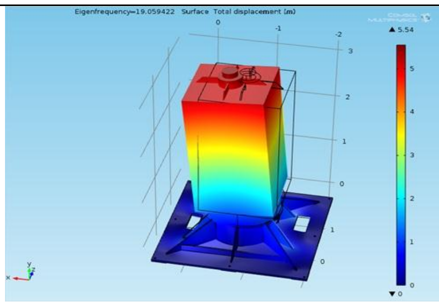

Figure **(f)**

Figure (8) “Modal analysis with comsol software for motor and foundation“

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MOD SHAPE

A Theoretical Model for pumping station by using Modal Shape program was done to define the high places stresses on the various parts of the motor and for motor foundation. Also Solid Work Program was used to drawing this pumping station to represent the problem in theory.

Dynamic Analysis (After Mode Shape)

Overall vibration level was measured again and the results were analyzed. The measurements results indicating that the ringing problem still exist and unresolved although, decreasing the level of overall vibration with almost about 40% but, it is still high on the motor where the level of vibration is reached to 8.9 mm/sec as shown in **"Table [2]"**. These levels are dangerous and not permissible according to ISO Standard.

Frequency Analysis (After Mode Shape)

Frequency spectrum analysis indicated that still vibration peaks are found at motor running speed 1RPM (16.5 Hz, 12.5Hz) and its harmonics reached 13.07 mm/sec as shown in **"Figure (9-a)"**, its harmonics as shown in **"Figure (9-b)"**, and peaks with high amplitude at low frequency as shown in **"Figure (9-c)"**.

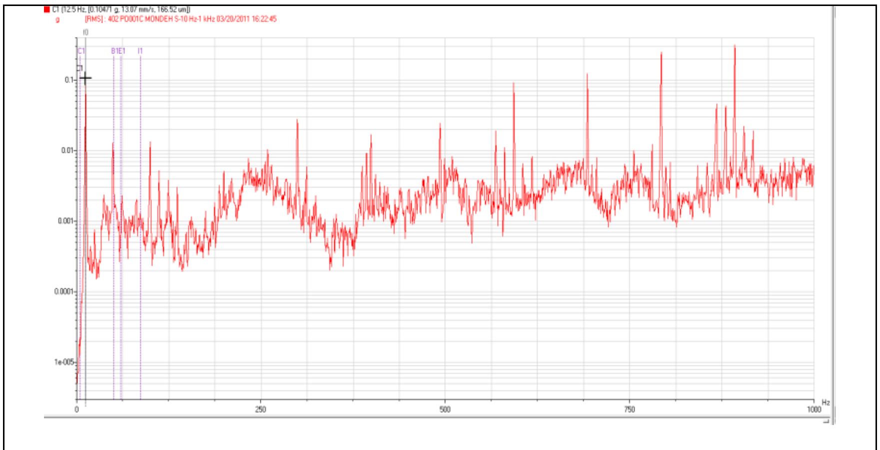

Figure (9-a) **"**FFT Domain for MNDEH**"**

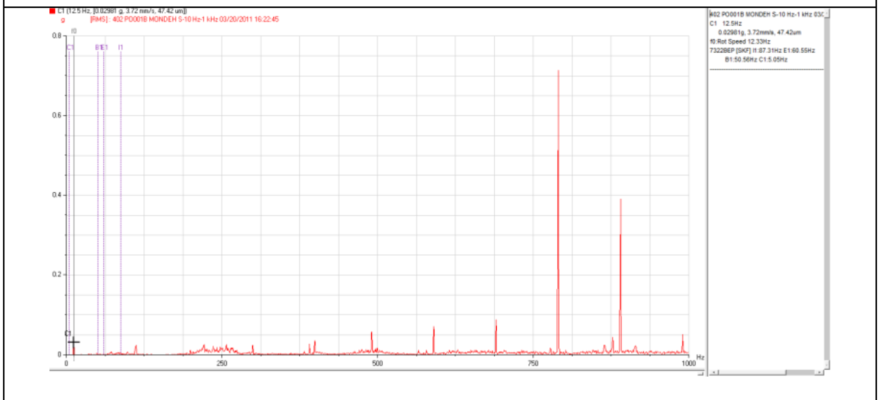

Figure (9-B) "FFT Harmonic for Resonance at MNDEH

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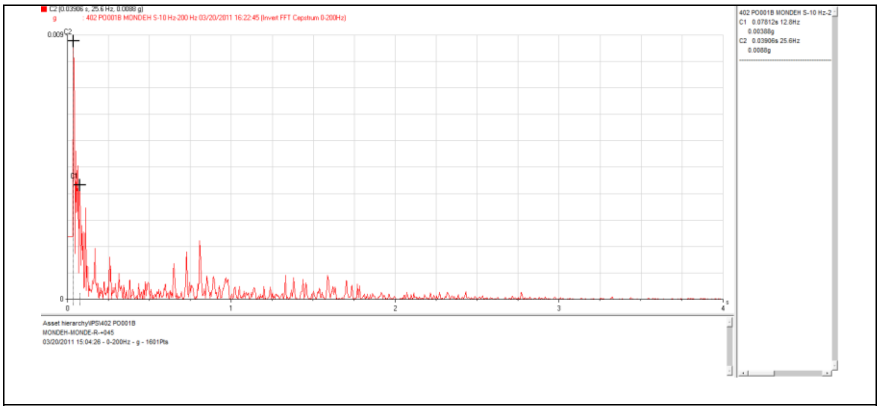


Figure (9-c) **“**Time domain for Peak at low frequency at MNDEH“

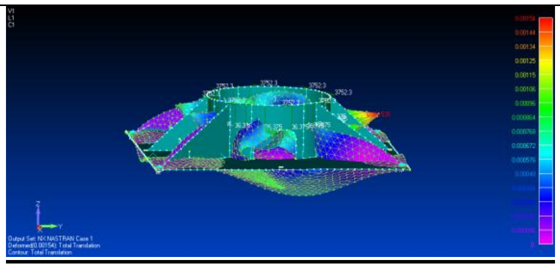
Figure (9) **“**Frequency Analysis (After Mode Shape) “

MODEL MODFFICATION

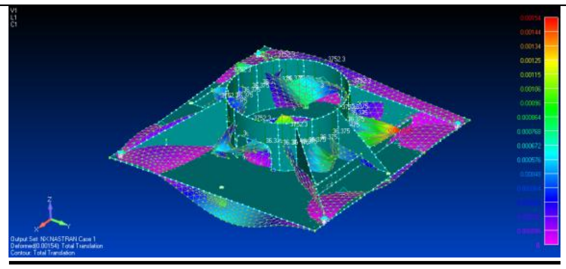
A new simulation was done again by the theoretical program.

Modified Model Results

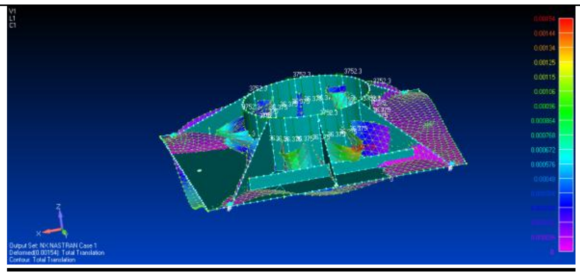
In this case the decision for the program was to increase Stainless steel supports for the motor foundation. Therefore, after increasing Stainless steel support of motor foundation for second time re-measuring were done for dynamic analysis and doing frequency analyses to determine the state of the system. Modal Shape for pump, motor, and foundation are shown in **Figure (10-a, 10-b, 10-c, 10-d, 10-e, and 10-f)**.



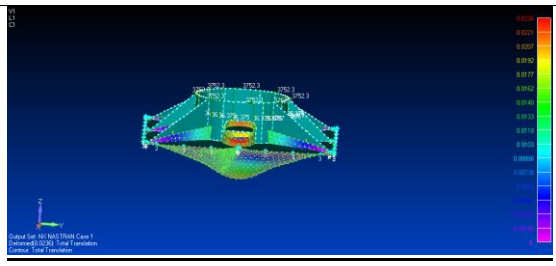
(a) finite element actual 1



(b) finite element actual 2



(c) finite element actual 3



(d) finite element front

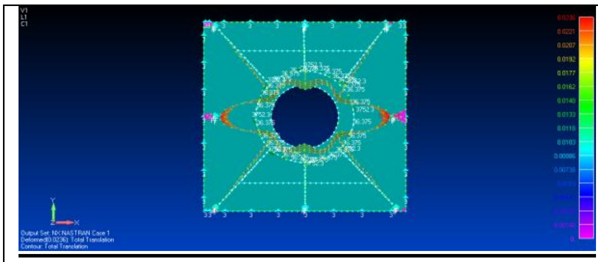
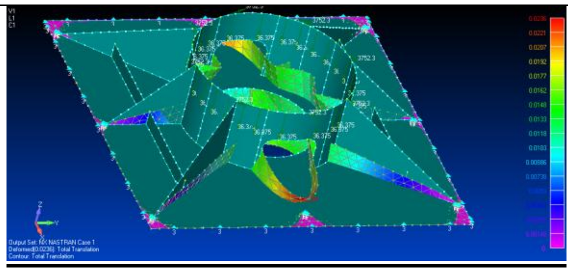
	
(e) finite element top	**(f)** finite element isometric2

Figure (10) **"**Modal analysis for foundation only with Ansys software version 14.5**"**

Dynamic Analysis (After Model Modification)

Overall vibration level was measured again after model modification and the results were analyzed. The measurements results indicating that the ringing problem was disappear and solved. The level of overall vibration is in the range of 3.9 mm/sec. The level of vibration improved with almost about 80%, as shown in **"Table [2]"**. These levels are acceptable according to ISO Standard.

Frequency Analysis (After Model Modification)

Frequency spectrum analysis indicated that no vibration peaks were found at motor running speed (1RPM, 12.5Hz, reached to 3.98 mm/sec, as shown in **"Figure (11-a)"**, no harmonics as shown in Figure (11-b), and no peaks with high amplitude at low frequency as shown in **"Figure (11-c)"**.

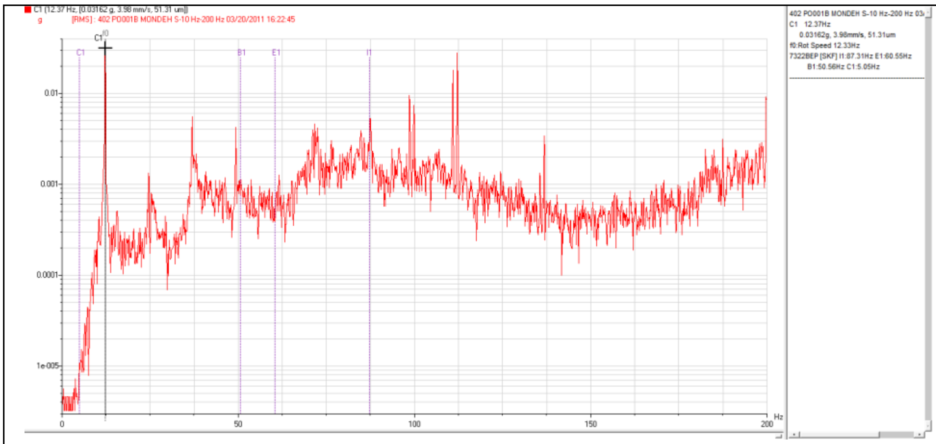


Figure (11-a) **"**FFT Domain for MNDEH**"**

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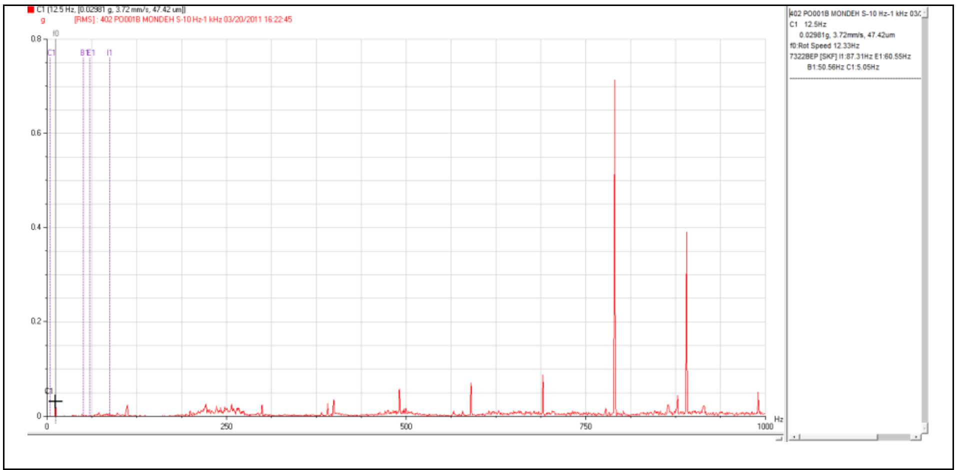

Figure "(11-B)" FFT Harmonic for Resonance at MNDEH"

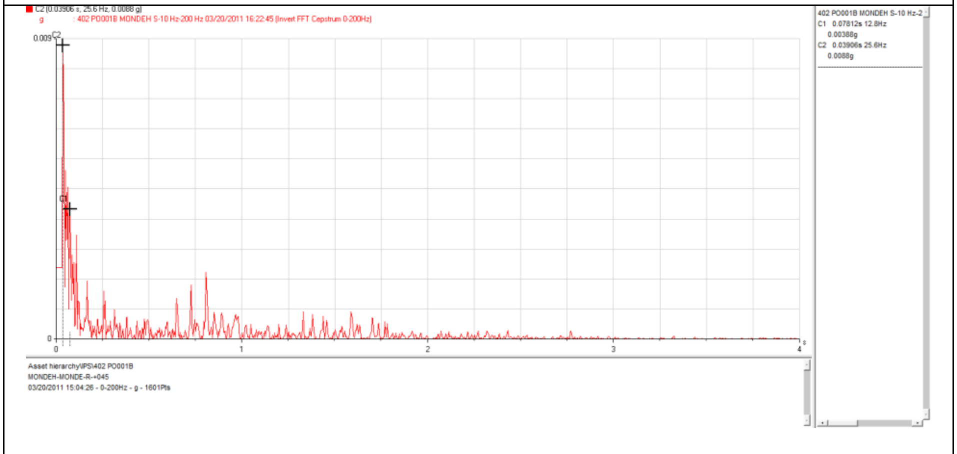

Figure (11-c) "Time domain for Peak at low frequency at MNDEH"

Figure (11) "Frequency Analysis (After Mode Modification) "

CONCLUSIONS

- Vibration problem of Helwan Irrigation Pumping Station is studied and analyzed to define vibration sources leading to damaging the foundation of most pumps in the station
- Experimental investigation was done and a finite element model for a vertical pump was built to solve vertical pumps structural weakness.
- High vibration levels were measured at 1x running speed of the motor due to resonance problem.
- The run-up tests confirmed a natural frequency at 16.5 Hz, which coincides with the motor running speed.
- The results of finite element analysis confirm that the third natural frequency is very close to 1x operating speed with deviation about 1%.
- The model suggests different scenarios to solve the structural problem and overcome the resonance by adding steel supports to the motor foundation
- The resonance problem solved by increasing the machine stiffness at the lower motor base by doing structural modifications.

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- The results confirmed that overall vibration level decreased 89% by structural modifications and the dynamic characteristics enhanced well.

REFERENCES

[1] Swarup, J. , (1989) "Vibration Analysis of Centrifugal Pump.", SV Sound and Vibration, Vol. 23, No. 9, pp. 24-28.

[2] Abdel-Rahman, S.M. , (1989) "Dynamic Analysis & Control of Irrigation and Drainage Pumping Systems in Egypt", Ph.D Thesis, Production Eng. & Mechanical Design Dept., Faculty of Eng., Menoufia University, Shibin El-Kom, Egypt.

[3] Sinha J, Rama Rao A. , (2005) "Vibration Diagnosis of Failure of Mechanical Coupling between Motor and Pump Rotors" International Journal of Acoustics and Vibration, Volume. 10, No. 2.

[4] DeMatteo T. , (2001) "Operational Deflection Shape and Modal Analysis Testing to Solve Resonance Problems", Consultant Emerson Process Management / CSI Division, CSI RBM University.

[5] Scheffer C. , (2008) "Pump Condition Monitoring Through Vibration Analysis", Pumps: Maintenance, Design, and Reliability Conference.

[6] Simmons G (1992) "Journal of Mechanical Energy Science" I Mech E, Vol.206, No.1.

[7] R.K.Biswas , (2006) "vibration based condition monitoring of rotating machines", national conference on condition monitoring [NCCM-2006] pg no 34-40.

[8] Department of the Army and Airforce, TM 5-805-4 /AF JMAN 32-1090, , (1995) *"Noise and Vibration Control,"* Tech. Manual, Deadquarters, Dept. of the Army and the Airforce, *Washington D.C.*.

[9] RGBS (Reservoir and Grand Barrage Sector, (2005) *"Zefta Rehabilitation/reconstruction Reversibility Study",* Summary Report, RGBS Tech. Workshop, *Cairo.*

[10] Doebling, S., et al., , (1996) *"Damage Identification and Health Monitoring of Structural and Mechanical systems from Changes in Their Vibration Characteristics,"* A Literature Review, Los Alamos National Laboratory, Report No. LA-13070-MS, Los Alamos, *New Mexico*.

[11] Rytter, A., , (1993) *"Vibration Based Inspection of Civil Engineering Structures,"* Ph.D Dissertation, Dept. of Building Technology and Structural Eng, Aalborg University, *Denmark.*

[12] Loh, C.H., et al., , (1995) *" A System Identification Approach to the Detection of Changes in Both Linear and Nonlinear structural Parameters*, Int. Journal Earthquake Eng. and structural Dynamics.

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