EFFECT OF INTAKE MAINTENANCE ON PUMP RELIABILITY AND PERFORMANCE

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ABSTRACT

The trend toward higher speed, higher power density liquid turbomachinery has inevitably increased the potential for fluid/structure interaction problems, and the severity of those problems. Fluid structure interaction phenomena can lead to increased wear and to structure failure. There are many factors affecting hydraulic and dynamic performance of pumping plants. These factors contribute to decrease performance efficiency, increase maintenance and replacement costs, and decrease reliability and operating life of the plants. Fortunately, vibration is a good indicator of pump performance and can early predict any failure or damage avoiding sudden breakdown. Through vibration monitoring, reliability and safety of pump plants can be ensured. Vibration analysis offers a comprehensive method of identifying a variety of pump problems.

Aquatic weeds and waste pollutants in the sump basin of pump plant block pump intake, reduce pump suction, and produce hydraulic problems affecting performance and efficiency of the plant. Hydraulic problems contribute significantly to downtime in pumping stations. Maintenance of pump intake is very important to ease water entrance to the plant, decrease consumed power, and avoid flow problems such as turbulence and eddies in the intake. In this research, dynamic and hydraulic performance was recorded on Al-Twesa Pump Station, Kom Ombo, Aswan. Heavy deposits and biological growth are found in the suction basin of the station. The primary results show that pump discharge is smaller than that designed and the vibration level is high. Amplitude of vibration is directly proportional to the amount of dynamic loads generated. Measurements were repeated after removing of aquatic weeds and cleaning the sump basin, and the results show that intake maintenance decreases vibration level from 25 % to 70 %. Hydraulic power increased 50 %, overall static efficiency reached up to 70 % as well as reliable and safe dynamic performance achieved. The research points out the importance of regular maintenance and cleaning of pump intake with regular vibration monitoring to early predict and diagnose pump faults, and maintain efficient pump performance and water requirements.

Keywords: Pump sump problems, Pump inlet conditions, Pump performance**,** Vibration

INTRODUCTION

Proper operation of pumping stations is greatly dependent upon the entire pump/piping system, which includes the piping geometry and the system operating conditions. Pumping stations are subjected to many operational problems affecting their performance and efficiency. It is very important to keep these stations efficient and reliable for irrigation and drainage purposes. Vibration has been recently adapted for condition based maintenance as a tool for early detection and diagnosis problems of pumping stations. Vibration measurements and analysis has proven to be the workhorse of machine condition assessment. Because of the intimate relationship between the shaft or casing vibration and the disturbing forces acting on the machine's internal components, vibration is a sensitive indicator of changes in machine condition which influences the fluctuating loads on machine components. Other operational or process parameters such as temperature, suction pressure, flow rate…etc., may yield significant information, but vibration is usually the most versatile and fundamental condition-related parameter. Vibration is used for evaluating running conditions of pumps as well as the need for maintenance or replacement works.

Causes of vibrations are of major concern because of the damage to the pump and piping that generally results from excessive vibrations. Vibrations in pumps may be a result of improper installation or maintenance, incorrect application, hydraulic interaction with the piping system, or design and manufacturing flaws. Hydraulic causes of vibration will be studied in this research. Wachel & Tison **[1]** list some of the common hydraulic causes of excessive vibrations and failures:

- Hydraulic instabilities
- Acoustic resonance (pressure pulsations)
- Water hammer
- Cavitation
- High flow velocity
- Operating off the best efficiency point of the pump.
- Impeller vane running too close to the pump cutwater.
- Internal recirculation
- Air getting into the system through vortexing….. etc.
- Turbulence in the system (non laminar flow).

Many of these causes are a result of an interaction of the pump with the fluid or the structure. This interactive relationship requires that the complete system should be evaluated rather than investigating individual components when problems occur. Improper inlet conditions cause suction recirculation that generated broadband turbulence. The turbulent energy excited acoustical resonance of the pump/piping system, resulting in pulsation at several discrete frequencies (Szenasi & Wachel **[2]**). This energy subsequently excited mechanical natural frequencies of the motor/pump/piping system causing high amplitude nonsynchronous vibration of the pump and other structures far downstream from the pump,

PUMP HYDRAULIC EXCITATION FORCES

The primary frequency of hydraulic excitations is the vane-passing frequency. The forces and the resulting vibration depend upon several factors, controllable at the design and specification stage. The most important method of minimizing vane-passing vibration is to design proper suction piping, to use double volute or centered volute pump designs wherever possible, and most important to operate the pump at or near the best efficiency point. For all types of volutes, it is possible to lower vane passing forces at some expense of efficiency by opening up the clearance between the impeller vanes and the volute tongue or diffuser vanes (Sanks et al., **[3]**).

In the pumps, the impellers that are the sources of noise and vibration define mainly the operation life. There are no other forces of mechanical nature generated by an impeller besides the centrifugal forces, defined by the imbalance. All the main oscillating forces are defined by the interaction of the flow with the blades of the impellers and with the inner surface of the case.

The main force applied to each blade of the impeller is the lifting force that is a result of the interaction of the flow with the blade that has a position with some angle against the flow. The sum of the lifting forces that act on an impeller without defects in a uniform flow is directed along the axis of its rotation and has no variable component. When the mean value or the direction of the lifting force of one of the blades differs form the others, the impeller will be exposed to first of all a radial to the rotation axis force with the rotating frequency and secondly, to a moment of forces with the same frequency, the vector of which will be also directed perpendicular to the rotation axis.

If the flow in the operating zone of the impeller is nonuniform then a radial force and a moment of forces of the same nature is applied to the impeller but the frequency of these forces is defined by the number of blades on the impeller. This frequency is called the blade-passing frequency (number of blades times shaft-rotating frequency). The described forces and moments are transferred via the rotation supports from the impeller to the case of the machine and simultaneously the pulsation of flow pressure impact the same case via the liquid or gas. The flow burble from the blade tips, and finally, the pulsation from cavitation in liquid in the pumps where this cavitation is present. The case vibration, excited by the variable pulsation of pressure in the flow, has a random nature and hence as no fixed frequency (Barkov **[4]**).

With certain flow conditions, piping systems will develop high levels of noise and vibration that can damage the pipes and related systems such as tube bundles, side cavities, and bluff or tapered bodies in flow streams. Pipe damage compromises plant safety, forces shutdowns, increases maintenance, and reduces efficiency and capacity. Fluid structure interaction phenomena can lead to increased wear and, under the worst conditions, to structural failure. It is recognized that the occurrence of these problems has contributed significantly to downtime in pumping stations (Abdel-Rahman **[5]**).

A large number of field problems have been experienced in the pump industry with both vertical and horizontal centrifugal pumps where the suction systems include sumps. The study and solutions of these cases, which included field-lab data and also theoretical investigation, have identified several specific hydraulic phenomena that can have negative impacts on the pumps performance and plant availability (Schiavello et al**.,** **[6**]). These phenomena, which must not be present to an excessive degree (ANSI/HI **[7]**) are:

- Free-surfaces vortices (originating at free water level)
- Submerged vortices (originating at solid boundaries, floor and back/side walls)
- Excessive preswirl of flow entering the pump
- Nonuniform spatial distribution of flow velocity at the impeller eye
- Excessive variation in velocity swirl with time
- Entrained air or gas bubbles

It has been recognized since the mid 1970s, (Bush et al**., [8]**) that operating pumps at reduced flows can generate harmful effects, such as high pressure pulsation, vibration, noise, and unsteady dynamic loads. Experimental investigations with centrifugal flow pumps using both flow visualizations (Minami **[9]**) and internal flow measurements (Schiavello **[10],** Sen **[11]**) have clearly shown that when the capacity is reduced below the best efficiency point, a complex three-dimensional flow pattern suddenly appears at the impeller inlet. This flow pattern is induced from the impeller itself and is characterized by:

- Flow reversal at the eye of the impeller (i.e., negative axial velocity component) also called "backflow"
- A vortex with tangential velocity swirling at the rotational speed also called "peroration"
- Radial static pressure distribution with higher value at the outer periphery (suction pipe wall)

PUMP INLET AND INTAKE PROBLEMS

Guidelines have been established for preventing pump hydraulic problems or minimizing the undesired effects (Flowserve **[12],** Claxton **[13]**). These guidelines include recommendations for sump designs, where critical dimensions are normalized with the suction bell inlet diameter, such as: clearances between the suction bell and floor side walls, sump width-length, baffles, and corner fillers. Additional recommendations are provided for operational parameters, such as minimum submergence, and velocity of the approach inflow across the sump and at the bell inlet.

Pump inlet conditions are among the most overlooked and misunderstood aspects of inlet design, yet they probably constitute the single reason most responsible for the success or failure of an installation. Pump performance is entirely dependent on the quality of the effort to provide adequate conditions at the pump inlet. Regardless of the type of inlet (either pressurized. sump. or forebay), care must be exercised to avoid poor hydraulic conditions at the impeller. The pump intake design must satisfy the requirements for proper approach conditions by avoiding the following (Sanks **[3]**):

- Loss of efficiency and capacity
- Noise and vibration
- Unstable operation
- Damage to impellers, bowls, bearings, and shafts.

The geometry of the pump sump and its associated piping and entrance conditions has a profound effect on pump performance. Hydraulic conditions detrimental to pump performance include (Sanks **[3]**):

- Asymmetrical velocities in the approach to the pump intake
- Nonuniform or asymmetrical velocity distribution in the pump intake
- Swirling or peroration of flow at the impeller.
- Surface and subsurface vortices.

Asymmetrical flow to a pump intake is likely to produce swirling and/or asymmetrical velocity distribution in the intake throat. Asymmetrical velocities in the intake throat create a higher load on one side of the impeller, bend the shaft, and put extra stress on bearings and couplings. They can cause rough operation, vibration, and loss of head and capacity. These effects worsen as pumps get larger and as the specific speed increases.

Swirling changes the angle of attack on the impeller vanes, reduces head and capacity, and decreases efficiency. An angle of 5° from axial is usually considered the maximum allowable. Swirling in the approach can degenerate into vortices. The pressure in the core of a vortex is reduced and air (or other gasses) coming out of solution can cause noisy operation and vibration. When a vortex is severe, it results in cavitation that quickly erodes metals.

Strong surface vortices entrain air bubbles that collapse as they pass from low to high pressure zones across the impeller vane, thus creating noise, uneven torque, undue stress on shafts and couplings, and erosion. A small amount of air can cause a large decrease of capacity, head, and efficiency. Milder surface vortices tend to become smaller but more intense as they enter the intake.

Strong subsurface vortices, formed as liquid separates from walls or floors, are often present and difficult or impossible to detect in the field. Normally, the bubbles in subsurface vortices consist of air that comes out of solution due to the decreased pressure in the core. Subsurface vortices are just as damaging as surface vortices-sometimes more so. If vapor bubbles can form, they cause rapid erosion. In fact, the rate of erosion can be reduced (although performance was also reduced) by introducing air deliberately to form large bubbles that do not completely collapse.

FLOW INDUCED VIBRATION

Liquid vibration sources are pressure fluctuations produced directly by liquid motion. Liquid vibration can be produced by vortex formation in high-velocity flow (turbulence), pulsations, cavitation, flashing, water hammer, flow separation, and impeller interaction with the pump cutwater. The resulting pressure pulsations and flow modulations may produce either a discrete or broad-band frequency component. If the generated frequencies excite any part of the structure including the piping or the pump into mechanical vibration, then vibration may be radiated into the environment. Four types of pulsation sources occur commonly in centrifugal pumps (Szenasi & Wachel **[2]**, Lobanoff **[14]**):

- Discrete-frequency components generated by the pump impeller such as vane passing frequency and multiples.
- Flow-induced pulsation caused by turbulence such as flow past restrictions and side branches in the piping system.
- Broad-band turbulent energy resulting from high flow velocities. Intermittent bursts of broad-band energy caused by cavitation, flashing, and water hammer.

Three different categories of flow oscillation can occur, and there are a number of phenomena within each of them. The three categories include: global flow oscillations, local flow oscillations, and radial and rotodynamic forces as follows (Brennen **[15]**):

[A] Global Flow Oscillations:

- Rotating stall or rotating cavitation occurs when a turbomachine is required to operate at a high incidence angle close to the value at which the blades may stall.
- Surge is manifest in a turbomachine that is required to operate under highly loaded circumstances where the slope of the head rise/flow rate curve is positive.
- Partial cavitation or super cavitation can become unstable when the length of the cavity approaches the length of the blade so that the cavity collapses in the region of the trailing edge.
- Line resonance occurs when one of the blade passing frequencies in a turbomachine happens to coincide with one of the acoustic modes of the inlet or discharge line.
- Cavitation noise can sometimes reach sufficient amplitude to cause resonance with structural frequencies of vibration.
- The above items all assume that the turbomachine is fixed in a non-accelerating reference frame. When this is not the case the dynamics of the turbomachine may play a crucial role in generating an instability that involves the vibration of that machine as a whole.

[B] Local Flow Oscillations:

- Blade flutter. There are circumstances under which an individual blade may begin to flutter (or diverge) as a consequence of a particular flow condition (incidence angle, velocity), blade stiffness, and method of support.
- Blade excitation due to rotor-stator interaction.
- Blade excitation due to vortex shedding or cavitation oscillations.

[C] Radial and Rotor Dynamic Forces:

- Radial forces are forces perpendicular to the axis of rotation caused by circumferential nonuniformities in the inlet flow, casing, or volute. The loads acting on the impeller and, therefore, the bearings can be sufficient to create wear, vibration, and even failure of the bearings.
- Fluid-induced rotordynamic forces occur as a result of movement of the axis of rotation of the impeller. Contributions to these rotordynamic forces can arise from the flow through the impeller, leakage flows, or the flows in the bearings themselves. Sometimes these forces can cause a reduction in the critical speeds of the shaft system, and therefore an unforeseen limitation to its operating range. One of the common characteristics of a fluid-induced rotordynamic problem is that it often occurs at subsynchronous frequency.

DESCRIPTION AND APPROACH OF THE CURRENT PROBLEM

Efficiency of the pumping stations constitutes the cornerstone of any maintenance and replacement works. Al-Twesa pumping station consists of 7 vertical pumps of axial flow type as shown in **Fig. (1)**, of 5.2 m^3/sec rated flow, 11.8 m static head, 845 kW electric power, and 996 rpm rotation speed. The station was constructed and began in 1968 for serving 40,000 feddans. Complete replacement of the pumps was done in 1998 and complete overhauls were done for the seven units from July 2001 until April 2002. Due to deficiency of the water pumped with respect to the designed flow rate and the high noise levels emitted from the units, it was required to evaluate the current efficiency of the station and evaluate the reliability of replacement and overhauls works done lately. Hydraulic and dynamic measurements were done in July 2002. Measurements include static head, flow rate, consumed power, and vibration analysis. The measurements show that the pumps don't provide the required flow rate and they operate away from the designed point (best efficiency point), as well as the vibration level is high according to the standards. Through dynamic analysis, the cause of high vibration level and inadequate flow rate was defined. The suction basin was full of aquatic weeds and Nile Rose, as shown in **Fig. (2)**. The mechanism of preventing weeds from entering the pump intake doesn't work and there is no enough workers in the station to clean sumps from weeds, sediments, fouling, and trees pollution, as shown in **Fig. (3)**. The problem was cleared and the suction basin and pump intakes were cleaned from such contaminants. Measurements were repeated in August 2002, where the pumps proved efficient hydraulic and dynamic performance, as shown in the next sections.

Vibration was measured and analyzed using spectrum analyzer model 2526 Bruel & kjaer (B&K), Denmark supported with machine monitoring software Model 7170 B& K. Vibration signals were recorded in terms of root mean square velocity on the motor and gearbox in nine locations in the axial and radial directions as follows:

Location (1): motor drive end axial,
Location (2): motor drive end radial,
Location (3): motor drive end radial perpendicular,
Location (4): Gear box-motor drive end axial,
Location (5): Gear box-motor drive end radial,
Location (6): Gear box-motor drive end radial perpendicular,
Location (7): Gear box-pump drive end axial,
Location (8): Gear box-pump drive end radial, and
Location (9): Gear box-pump drive end radial perpendicular

RESULTS AND ANALYSIS

Vibration measurements and dynamic analysis were done on the seven units of Al-Twesa Pumping station to determine overall vibration levels and vibration spectra. Overall vibration levels which indicate severity of vibration are compared with ISO 10816-1 (Mechanical Vibration- Evaluation of machine vibration by measurements on non-rotating parts- part 1: General Guidelines). Vibration spectra, relation of vibration amplitude with frequency, are measured to determine the excitation frequencies and the source of high vibration. According to ISO 10816-1, class III zone B was used as acceptable limit for the pumping units. The good vibration limit is 1.8 mm/s vibration velocity, where acceptable limit is up to 4.5 mm/s. Hydraulic calibration for the seven units were done separately for each unit by measuring static head, flow rate, and consumed power. Overall static efficiency was calculated for each pump unit. Measurements were done in July 2002, where the suction basin of the station and the pump intakes were blocked of weeds, sediments, Nile Rose, solid pollutants and contaminants. The tests were repeated in August after cleaning the suction basin and the pump intakes.

1. Overall Vibration Levels

Overall vibration levels measured before intake maintenance show that vibration level is high in all seven units. The levels are different at the different nine measurement locations predescribed. Locations 1, 2, 3 are measured on the motor drive end in the axial, radial, and radial perpendicular respectively. Locations 4, 5, 6 are measured on the gear box - motor drive end in the axial, radial, and radial perpendicular respectively. Where Locations 7, 8, 9 are measured on the gear box - pump drive end in the axial, radial, and radial perpendicular respectively. The results show that the highest vibration level is measured on the motor drive end for all pumping units. This is due to the motor of vertical pump is working as a cantilever fixed at the end (pump) and free at the top (motor).

Overall vibration level for pumping unit (1) before and after intake maintenance is shown in **Fig. (4)**. The maximum vibration level measured at the motor drive end in the radial perpendicular direction before intake maintenance is 5.6 mm/s. This level is beyond the acceptable standard level; however the level was decreased to 2.6 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable and within good zone after intake maintenance. The results show that vibration level decreased from 30 % to 50 % in all locations after intake maintenance.

Overall vibration level for pumping unit (2) before and after intake maintenance is shown in **Fig. (5)**. The maximum vibration level measured at the motor drive end in the radial direction before intake maintenance is 5.2 mm/s. This level is outside the acceptable standard level; however the level was decreased to 2.6 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable after intake maintenance. The results show that vibration level decreased from 30 % to 50 % in all locations after intake maintenance.

Overall vibration level for pumping unit (3) before and after intake maintenance is shown in **Fig. (6)**. The maximum vibration level measured at the motor drive end in the radial perpendicular direction before intake maintenance is 4 mm/s. This level is within the acceptable standard level and the level was decreased to 2.8 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable before and after intake maintenance. However, vibration level decreased from 25 % to 60 % in all locations after intake maintenance.

Overall vibration level for pumping unit (4) before and after intake maintenance is shown in **Fig. (7)**. The results show the same trend as unit (3). The maximum vibration level measured at the motor drive end in the radial perpendicular direction before intake maintenance is 4.4 mm/s. This level is within the acceptable standard level and the level was decreased to 2.0 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable before and after intake maintenance. However, vibration level decreased from 25 % to 60 % in all locations after intake maintenance.

Overall vibration level for pumping unit (5) before and after intake maintenance is shown in **Fig. (8)**. The maximum vibration level measured at the motor drive end in the axial direction before intake maintenance is 4.1 mm/s. This level is within the acceptable standard level and the level was decreased to 2.0 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable before and after intake maintenance. However, vibration level decreased from 25 % to 50 % in all locations after intake maintenance.

Overall vibration level for pumping unit (6) before and after intake maintenance is shown in **Fig. (9)**. The maximum vibration level measured at the motor drive end in the axial direction before intake maintenance is 4.0 mm/s. This level is within the acceptable standard level and the level was decreased to 2.6 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable before and after intake maintenance. However, vibration level decreased from 35 % to 50 % in all locations after intake maintenance.

Overall vibration level for pumping unit (7) before and after intake maintenance is shown in **Fig. (10)**. The maximum vibration level measured at the motor drive end in the radial perpendicular direction before intake maintenance is 3.6 mm/s. This level is within the acceptable standard level and the level was decreased to 1.0 mm/s after intake maintenance. The vibration levels measured on all locations are acceptable before and after intake maintenance. However, vibration level decreased from 40 % to 70 % in all locations after intake maintenance.

2. Dynamic Analysis

Dynamic analysis was done to record the vibration spectra at different suction sump conditions to determine exciting frequencies, and evaluate sources of high vibration. Vibration spectra, shown in **Fig. (11)**, were measured before intake maintenance for pumping units (1), (7). The spectra show some minor mechanical problems of small vibration level measured at the motor speed (16.5 Hz), pump speed (6 Hz) and their harmonics. These problems include unbalance of the motor fan and misalignment of the shaft between motor and pump. These mechanical problems cannot be avoided at all. There should be some minor mechanical problems even for new or replaced pumps as long as they operate due to clearance and rubbing between rotating elements, and improper assembly. The mechanical problems can be decreased by regular maintenance and periodic inspection. All mechanical problems including unbalance, misalignment, bent shaft, bearings, and gearbox are diagnosed at some specified frequencies function of the rotation speed of the elements, components, and their geometry. On the other hand, hydraulic problems do not occur at specified frequency, but they occur at non-harmonic high frequency beyond 1 kHz. Vibration peak was measured at blade passing frequency (30 Hz) in all spectra indicating hydraulic problems. The measured spectra show high non-harmonic frequencies (≈ 3 kHz) on the shape of a dome indicating recirculation and turbulence hydraulic problem. The hydraulic problems generate dynamic loads of 0.5 mm/s vibration amplitude. Recirculation and flow turbulence in the suction basin was developed due to aquatic weeds and solid contaminants. The flow recirculation generates broadband turbulence of random, high amplitude, non synchronous vibration over a large frequency range. A vibration peak was also measured at frequency 5 Hz, which probable is the pump motor lateral natural frequency that was excited by the broadband energy produced when the pump was operating in circulation condition due to pollutants.

Dynamic analysis was repeated after intake maintenance and the spectra are shown in **Fig. (12)** for some pumping units. It is apparent that these spectra are different from those measured before intake maintenance. There is no more a dome shape in the high frequency range. Vibration peaks measured at the rotation speed of the motor and pump decreased more than 50 %. Blade passing frequency was not recorded in all pump units indicating free hydraulic problem in these pump units. There is no vibration peak at the low frequency range in the seismic range implying no structural resonance problem as before. Maintaining and cleaning the pump intake clear the aquatic weeds and sediments in the suction sump avoiding resonance problem that was generated before intake maintenance. The spectrum measured on pumping unit (7) show small vibration amplitude at low frequency and high frequency range identifying small mechanical and no hydraulic problems leading to safe and reliable pump unit. So, cleaning the intake affects dynamic performance of the seven pump units in different way. Some units improved dynamically more than others. Waste solids may take time to be removed completely and they still affect some units. Intake maintenance and cleaning should be done regularly. More results are found in the Technical Report (MERI, **[16]**)

3. Hydraulic Performance

Hydraulic performance for the pump units before intake maintenance shows that static head, flow rate, and overall static efficiency in all pumping units are lower than the designed point. The pump station does not fulfill the required water for irrigating of expected served area. However, the consumed electric power is less than rated due to low discharge pumped. Pump best operating point is 5.2 m^3/sec flow rate, 11.8 m static head, and 845 kW electric motor power. Static head measured is 9.5 m, flow rate is 2.8 m^3/sec, and overall static efficiency is in the range of 40. Hydraulic performance was repeated after intake maintenance and the results are shown in Table (1). Average flow rate measured ranges from 3.0 to 4.0 m^3/sec for the different pumping units, where average static head measured is 12.92 m. The measured static head is higher than designed due to lower water level in the suction basin. This high static head leads to lower flow rate. However, the results show that the hydraulic performance of the pumping units improved greatly after intake maintenance. Hydraulic power, flow rate times head, increased 50 %. Electric power consumption increased and overall static efficiency of the units reached to 70 % for some pumping units as well as reliable and safe dynamic performance attained. Pump unit (7) shows reliable and efficient hydraulic and dynamic performance. Maintenance pump station suction basin affects hydraulic performance of the seven units differently. Some units were hydraulically more improved than others. The hydraulic and dynamic performance were correlated well for the different pumping units implying more intake maintenance should be done regularly to obtain efficient hydraulic performance for all the pump units.

CONCLUSIONS

From the above, the following points can be concluded:

- Intake maintenance decreases vibration level up to 70 % for the different pumping units and vibration levels measured are within acceptable limit according to ISO 10816-1 after intake maintenance.
- Recirculation and flow turbulence in the suction basin was developed due to aquatic weeds and solid contaminants generating broadband random, non synchronous vibration over a large frequency range of 0.5 mm/s dynamic load amplitude.
- Pump motor lateral natural frequency was excited by the broadband energy produced by circulation condition.
- Hydraulic performance for the pumping units before intake maintenance shows that the pump station does not fulfill the required water for irrigation.
- Pumping operation is greatly affected by aquatic weeds and sediments in the suction basin. The results show that the hydraulic performance of the pumping units improved greatly after intake maintenance. Hydraulic power increased 50 %. Overall static efficiency for some pump units reached to 70 % as well as reliable and safe dynamic performance achieved.
- Maintaining pump station suction basin affects performance of the pumping station. The results of hydraulic and dynamic performance are correlated well for the different pump units implying more intake maintenance should be done regularly to improve pump performance.

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Table (1) Hydraulic Performance After Intake Maintenance

Unit No.	**Total Head (m)**	**Discharge (m^3/sec)**	**Electric Power (kW)**	**Overall Efficiency (%)**
1	12.92	3.653	732.36	63.2
2	12.92	3.026	732.36	52.3
3	12.92	3.579	739.93	61.3
4	12.92	3.752	741.8	64.1
5	12.92	3.993	732.86	69
6	12.92	3.58	700.22	64.8
7	12.92	3.932	705.22	70.6

Figure (1) Photograph of Al-Twesa Pumping Station

Figure (2) Aquatic Weeds in the Suction Basin

Figure (3) Photographs of Different Contaminants blocking Pump Intake

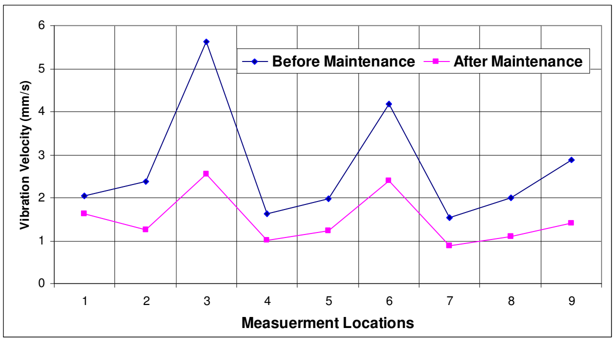


Figure (4) Overall vibration level before and after maintenance for unit (1)

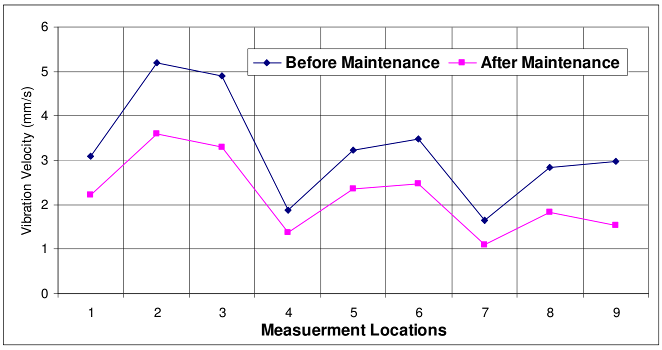


Figure (5) Overall vibration level before and after maintenance for unit (2)

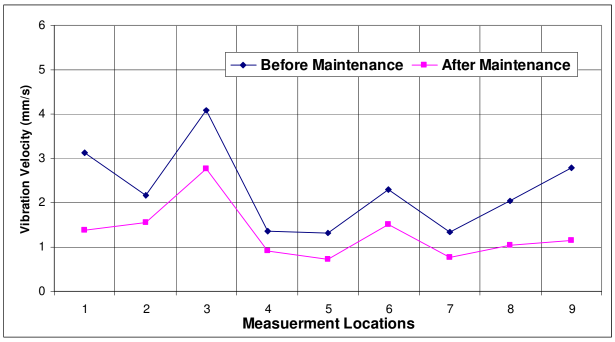


Figure (6) Overall vibration level before and after maintenance for unit (3)

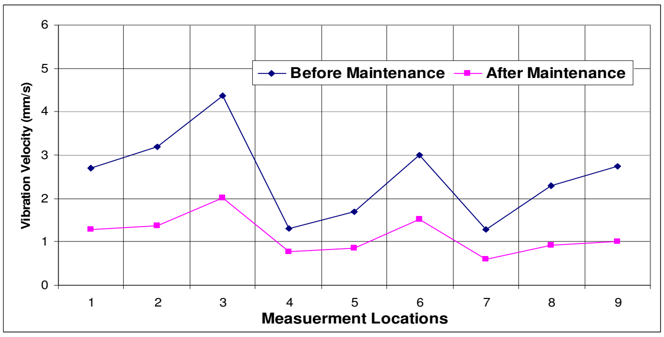


Figure (7) Overall vibration level before and after maintenance for unit (4)

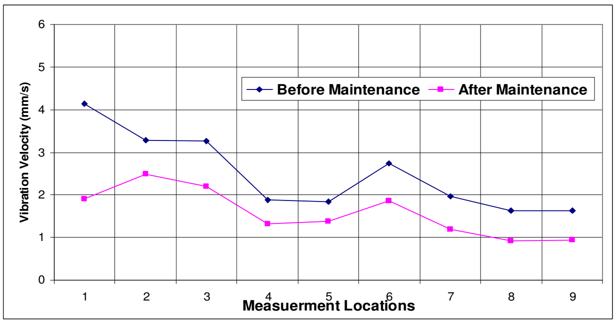


Figure (8) Overall vibration level before and after maintenance for unit (5)

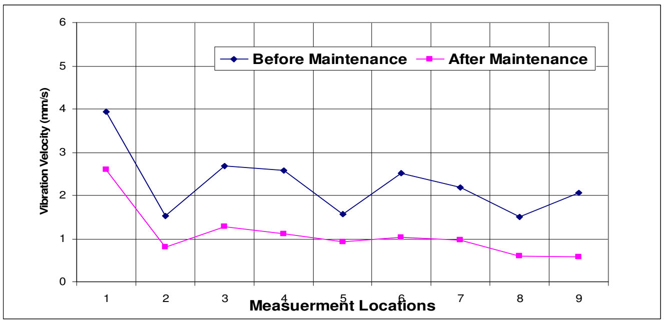


Figure (9) Overall vibration level before and after maintenance for unit (6)

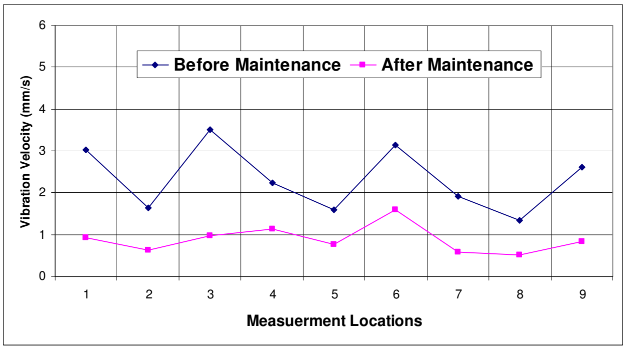
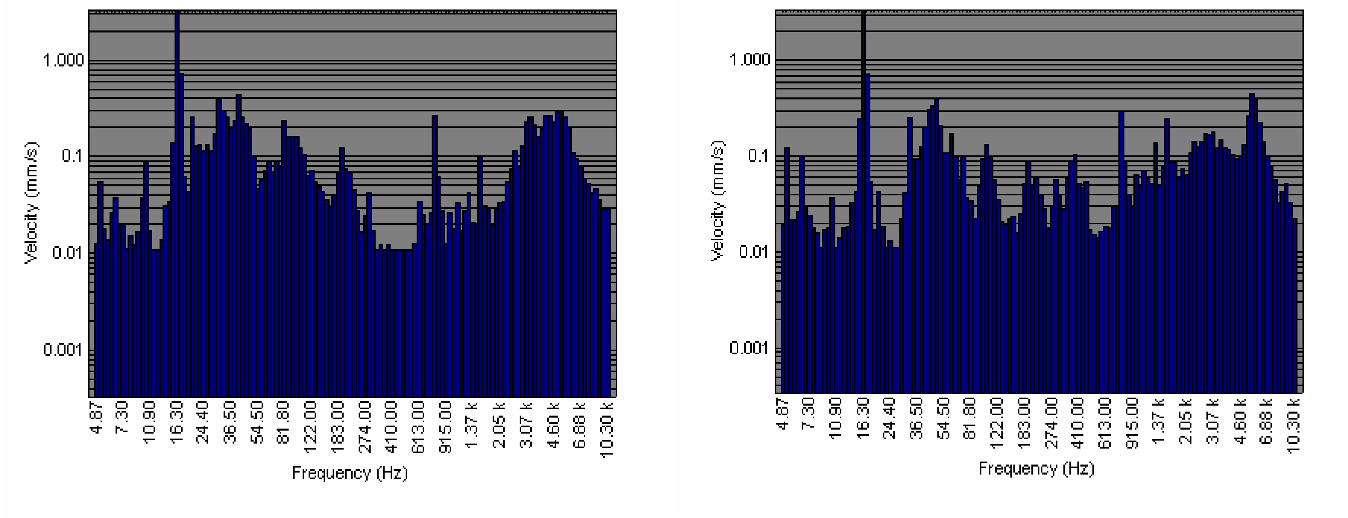
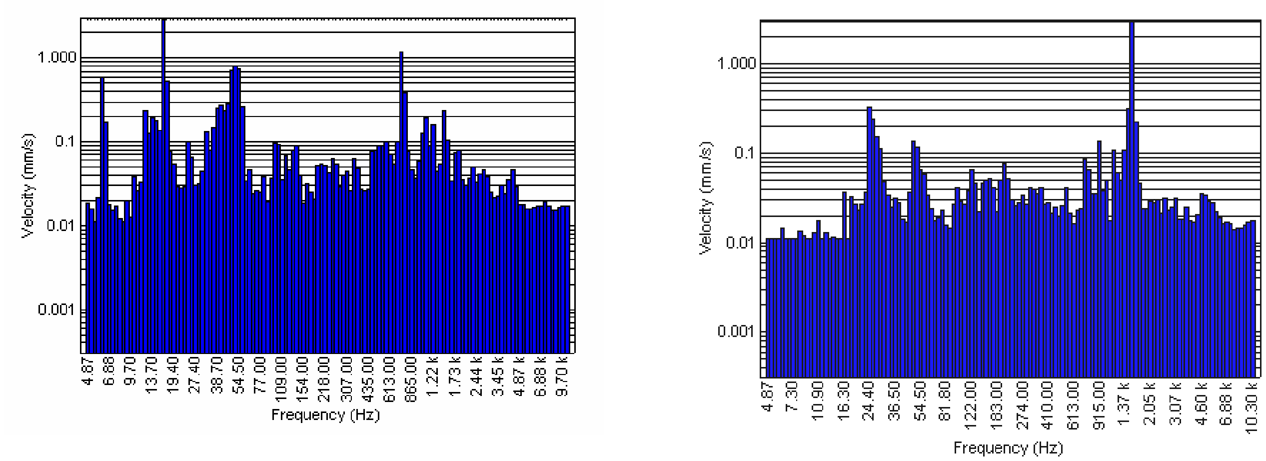


Figure (10) Overall vibration level before and after maintenance for unit (7)



Unit [1] **Unit [7]**

Figure (11) Vibration Spectra Before Intake Maintenance



Unit [1] **Unit [7]**

Figure (12) Vibration Spectra measured for different Pump Units After Intake Maintenance