

## VIBRATION ANALYSIS OF CENTRIFUGAL PUMP WITH VARIABLE SPEED DRIVES

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Variable speed drives (VSD) improve operation and performance of pumping system and allowing them to operate efficiently, match fluctuations in the water demand and hence saving energy over the required range of field operation. Variation of speed may considerably lead to coinciding of stationary speed with the critical speed. For VSD applications, the excitation frequencies become variable and the likelihood of encountering a resonance condition within the continuous operating speed range is greatly increased. Vibration tests were done to obtain the optimum Dynamic performance and its sensitivity due to speed variation. The present work was done on a test rig of a centrifugal pump unit with a variable speed drive. Forced vibration measurements were done at different operating speed ranged from 25 Hz to 50 Hz to determine vibration levels and exciting frequencies. Experimental modal testing was done at different conditions to simulate the design and installation cases. Operational modal analysis is done to simulate the actual operating conditions. Dangerous frequencies are well determined to avoid operation at resonance. Finite element model was done using ANSYS software to obtain the dynamic characteristics of the pump model analytically.

**KEYWORDS:** variable speed pumps, vibration analysis, modal analysis, resonance

### 1. INTRODUCTION

In most of water pumping stations, the speed of the pumping system is approximately constant with controllable flow rate to match the load demand. This water load demand is fulfilled by the traditional control methods such as throttling valves or bypass techniques to change the operating condition in the pumping system by increasing the system's backpressure or resistance of flow. This increase in pressure head shifts the pump's operating point to the left along its performance curve, and, typically, away from its best efficiency point. It is confirmed however, that these methods rise energy losses and hence bad efficiency performance over the range of operation [1]. It should be admitted that a variety of approaches can be considered to eliminate the traditional

control losses and reduce the electrical energy consumption of water pumps. It is believed now that the most appropriate and most economic way for improving the water pump performance is the Variable Speed Drives (VSDs) [2]. Pumping units when driven by (VSDs), will have a flexible adjustment and monitoring of pressure and flow over a very wide range of operation. In addition the VSD will improve the performance of pumping unit and allow the pump to operate as near as possible to its maximum efficiency and hence saving energy over the required range of field operation. (VSDs) adjust motor speeds driving the pump to match fluctuations in flow. In addition to energy savings, (VSDs) offer precise speed control and a soft-starting capability. Each control technique of course has its merits and demerits, so (VSDs) also have some potential drawbacks which can be avoided with appropriate design and application. Resonance problem is the most seriously drawback of the (VSDs) [3]. It can cause excessive vibration levels, which in turn are potentially harmful to equipment and environment. For (VSDs) applications, the excitation frequencies become variable and the likelihood of encountering a resonance condition within the continuous operating speed range is greatly increased from a variety of sources they may not have been excited by a constant speed drive [4]. The risk of the rotating element encountering a lateral critical speed increases when using (VSDs) [5]. However, vibration level increases with increasing speed of (VSDs) and amplitude of vibration increases greatly with increasing excitation forces. At normal operating condition, vibration level increased 3.8% by increasing running speed 66% [6]. It is known that varying speed of the pump affects head, discharge and power according to the affinity law relationships:

- Flow rate  $\propto$  speed :  $Q \propto N$
- Head  $\propto$  (speed)<sup>2</sup> :  $H \propto N^2$
- Power  $\propto$  (speed)<sup>3</sup> :  $P \propto N^3$

In centrifugal pump applications with no static lift, system power requirements vary with the cube of the pump speed. Small decreases in speed or flow rate can significantly reduce the used energy. For example, reducing the speed (flow) by 20% can reduce input power requirements by approximately 50% [7]. The objective of this research is to obtain the optimum dynamic behavior and its sensitivity due to speed variation of the variable speed pumping unit. The present work was done on a test rig of a centrifugal pump unit with variable speed drive in the hydraulic lab of Mechanical & Electrical Research Institute (MERI).

## 2. RESEARCH METHODOLOGY AND FACILITIES

The test rig consists of a centrifugal pump used for irrigation purposes in the hydraulic lab of Mechanical & Electrical Research Institute (MERI). The pump is of type Epara with power of 7.5 Hp, 9 A, 3000 rpm, flow rate of 36 m<sup>3</sup>/hr, and head of 7.5 m connected with variable speed motor. The pumping system consists of motor, pump, suction, and delivery pipelines.

Tests were done to evaluate dynamic performance of the pump unit. Forced vibration test is done to determine vibration levels and exciting frequencies using a dual channel (01dB Movipack analyzer) and a machine monitoring software package (eDiag) for evaluating dynamic running condition of the pump unit.

Experimental modal tests were conducted using a spectrum analyzer (B&K pulse 3560 hardware module 5 channels), Impact hammer to excite the pump unit with an impulse, accelerometer, and modal analysis software package (Modal Test Consultant — Type 7753 ). The modal testing facility is capable of determining the inherent dynamic characteristics of the pump unit including natural frequencies, damping factors, and mode shapes. Experimental modal analyses were done on the pump unit in two boundary and supporting conditions to simulate the design and installation conditions.

Operational modal analysis was conducted using LMS Test. Lab. Software with the (B&K pulse 3560 hardware module 5 channels) analyzer. The system is capable of determining natural frequencies, damping factors, and mode shapes which are important in evaluating the dynamic behavior of the system. Configuration patterns of the system at the resonant frequencies and the operating conditions were defined. The natural frequencies of the pumping system were defined and when the exciting frequency approaches the natural frequency of the test object, the vibration level at the resonance are rapidly approaches a sharp maximum value. The pumping system and all facilities are shown in Fig. 1.

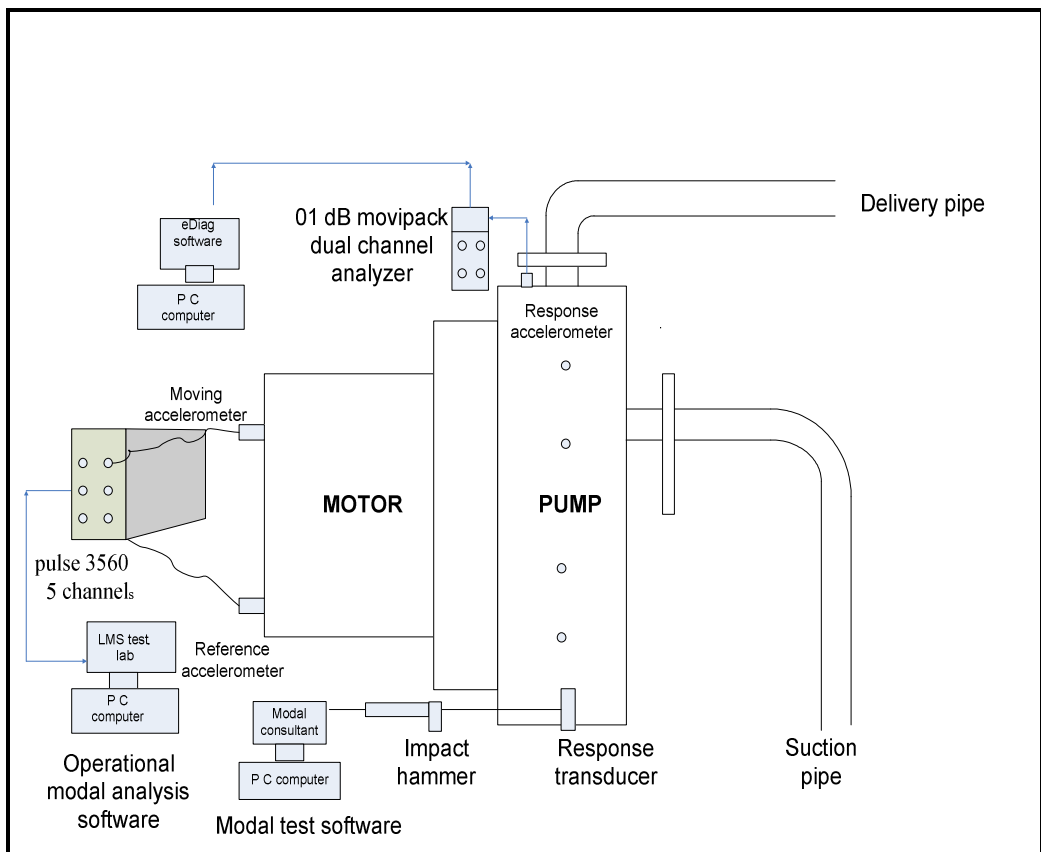


Fig. 1 Test facilities

### 3. FORCED VIBRATION ANALYSIS

Forced vibration analysis was done on the variable speed centrifugal pump unit at different speeds in the range (25, 30, 35, 40, 45, and 50 Hz) to specify the sources of vibration and define the exciting operational frequencies at different locations and different boundary conditions. The measurement location were done on the parts of the pumping unit including the motor and the pump at 8 locations in the axial, vertical, and radial directions as shown in Fig. 2. The measured parameters at every location are simple spectrum vibration velocity (FFT) in a frequency range from 2 Hz to 1 kHz and overall vibration velocity 10 Hz 2 kHz. The signals from the accelerometers are directly fed into the 01dB movipack analyzer which possesses an internal signal conditioning system comprising filters, integrators, amplifiers, etc. The signals are then transferred to the PC via USB connection to the eDiag software for signal analysis.

All these measurements were taken according to the ISO 1-10861 where the root mean square (RMS) was measured for all parameters because it is the most accommodate one for machine diagnosing where it expresses for the energy consumption due to vibration. It also takes the time history of vibrations [8]. The measurement locations are as follows:

- Point (1) motor non drive end in the radial direction.
- Point (2) motor non drive end in the vertical direction.
- Point (3) motor drive end in the radial direction.
- Point (4) motor drive end in the vertical direction.
- Point (5) pump drive end in the radial direction.
- Point (6) pump drive end in the axial direction.
- Point (7) pump drive end in the vertical direction.
- Point (8) pump non drive end in the horizontal direction.

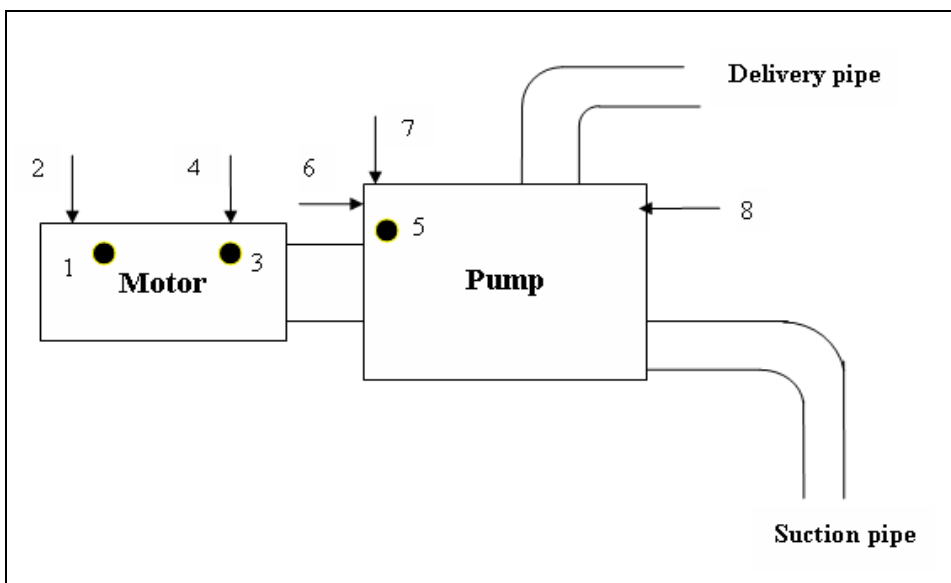


Fig. 2 Locations of vibration measurements

The measurements showed that the maximum overall vibration level measured was reached 1.58 mm/s on the motor and 1.28 mm/s on the pump at running speed 50 Hz, 1.1 mm/s on the motor and 1.06 mm/s on the pump at running speed 45 Hz, 0.815 mm/s on the motor and 0.646 mm/s on the pump at running speed 40 Hz, 0.646 mm/s on the motor and 0.594 mm/s on the pump at running speed 35 Hz, and 0.393 mm/s on the motor and 0.50 mm/s on the pump at running speed 25 Hz .

Vibration levels measured are good and safe according to ISO 1-10861 at all tests speeds except at resonance speed frequency (30 Hz), the overall vibration level reached value 2.91 mm/s while the spectra shows that the vibration amplitude reached a value of 3.07 mm/s at 30 Hz as shown in Fig. 3 and this value are dangerous and not permissible. The results proved that operation at resonant frequency generates dangerous vibration level. The results showed that vibration level increases slightly and smoothly with increasing running speed as excitation frequencies may be excited at higher speeds as shown in Fig. 4. The results indicated the importance of defining resonant frequencies firstly to avoid operation at these speeds or modifying structural properties to change resonant frequency through modal analysis.

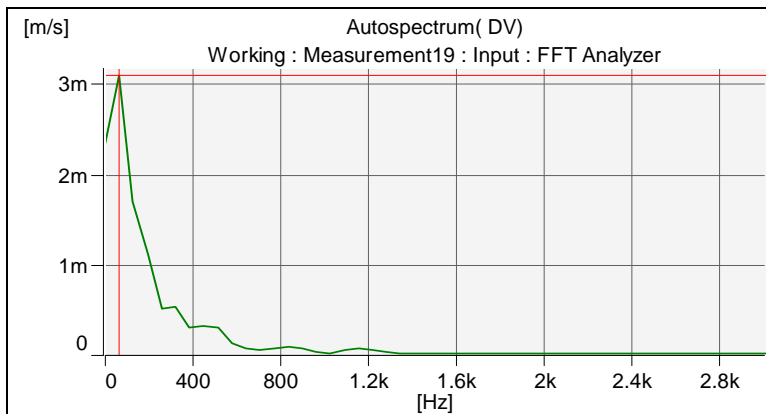


Fig. 3 Vibration spectrum measured at speed 30 Hz

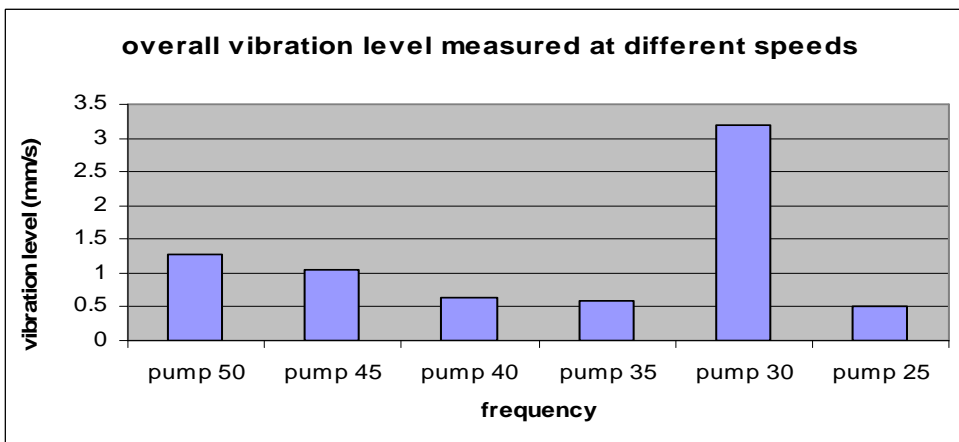


Fig. 4 Vibration level measured at different speeds on pump

#### 4. EXPERIMENTAL MODAL ANALYSIS

Modal analysis is a process whereby we describe a structure in terms of its natural characteristics which are the frequency, damping, and mode shapes which are called the dynamic properties. These properties are used to form the model of the dynamic behavior of structure system. Modal analysis is based on the fact that the response of a linear dynamic system can be expressed as the linear combination of a set of a simple harmonic motion called the natural modes of vibration.

Modal testing measurements were done on the whole pumping system at different boundary conditions at the laboratory. Different tests were done using different excitation and response locations to get reliable and applicable data. Experimental modal analysis was done at different phases of the pumping system representing design and installation stages. Modal testing was done by impacting the structure with the hammer to excite the system at fixed location and moving the response accelerometer around the whole structure at 47 locations as shown in Fig. 5.

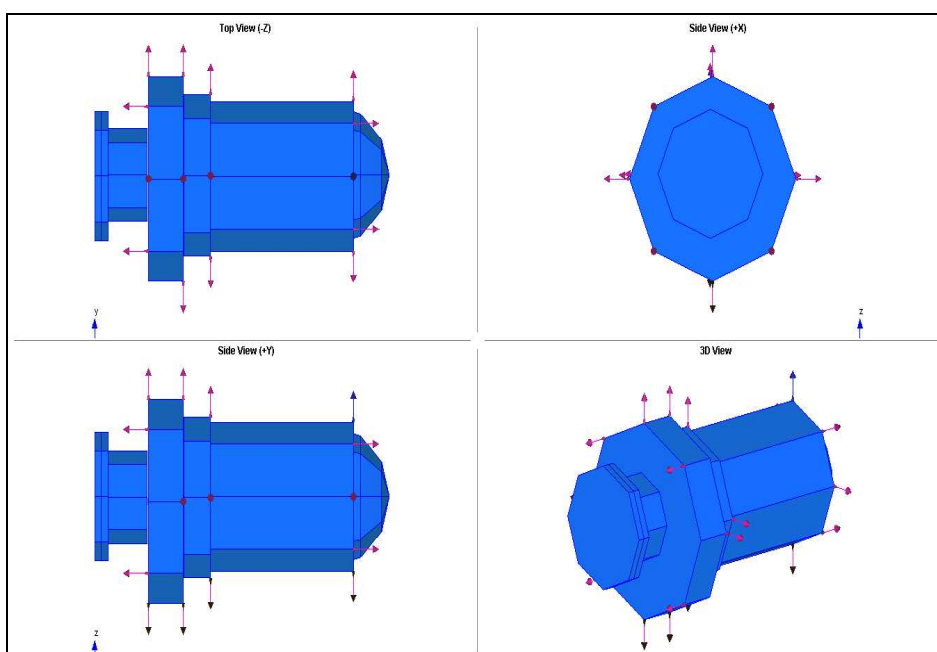


Fig. 5 Top and side views of the pumping system showing measurement locations

The experimental modal analysis was done on the pump unit in two boundary and supporting conditions to simulate the design and installation cases conditions. The pumping unit was first free support in air, and second was grounded support with pipe lines in water. Testing the structure in free condition determines the rigid body modes and thus the mass and inertia properties and it is recommended to always test structures in a freely supported condition. Also, testing the pump when it was grounded support in water with the pipe lines is to simulate the actual operation and installing conditions. Frequency Response Functions (FRFs) were measured for the pump unit free support in air, and for grounded support in water are shown in Fig.6 and 7.

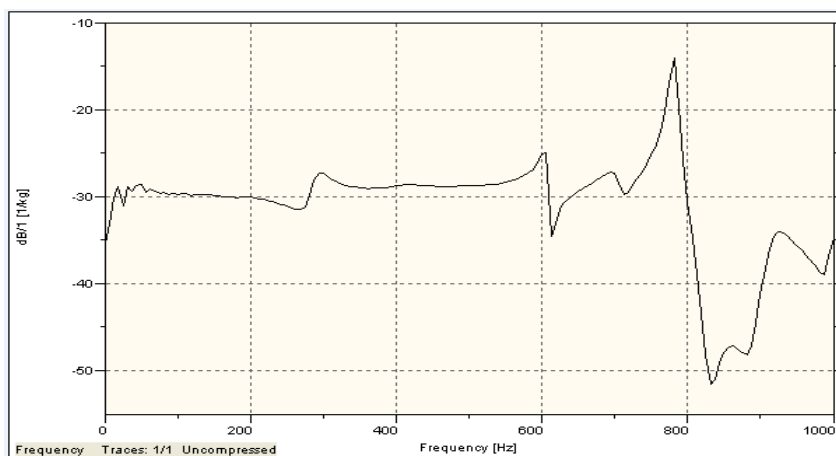


Fig. 6 FRF measured for the pump unit free support in air

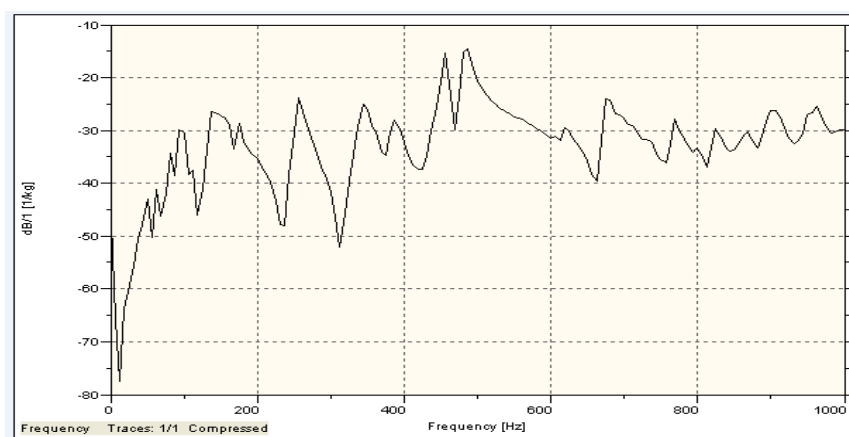


Fig. 7 FRF measured for the pump unit grounded support in water

Modal parameters extracted from FRFs measurements are located in Table 1. The results confirmed that there is a distinct coincidence between the natural frequency and the pump speed at 30 Hz, which indicated that the speed of 30 Hz is a critical speed. So, operation at resonant frequency generates dangerous vibration level.

The results indicated that the modal frequencies are different for the two cases as shown in Fig. 8. Some modes disappeared or shifted to other values. This is due to the different nature of each case, different fixation, different components, and complexity of the system. Also pumping system consists of many different items of different materials and properties when connected together or joined to other components will result in different dynamic behavior. The excitation at the pump casing will not have the same FRFs characteristics on the pump casing as well as the pipe lines for the above reason. The different components of the system have its separate modal properties called local modes that are when the system is integrated, the modal characteristics will be different from each single item forming global modes which may contain local modes of separate components.

**Table 1 Modal parameters measured from experimental modal testing**

Mode No.	Pump unit freely support in air	Pump unit ground supported in water
	Natural frequency (Hz)	Natural frequency (Hz)
1	4.9	15.24
2	23.80	30.0
3	37.70	68.40
4	130.12	81.20
5	260.60	86.0
6	307.22	122.40
7	833.51	130.0
8	858.57	200.0
9	867.54	248.50
10	900.18	265.36

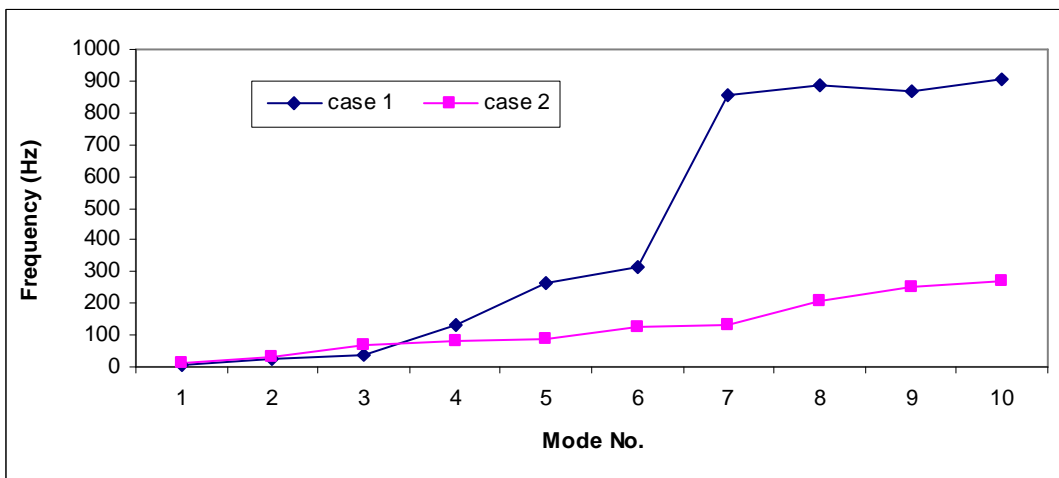


Fig. 8 Measured modal frequency changes for different two cases

Doing modal tests during design phase are very important to make sure that the harmonic and sub-harmonics of the exciting components of the pumping system (impeller, blades, bearings, rotating speed, etc..) are far from the natural frequencies of the pumping unit. Modal tests in installation phase is very important to make sure that pumping system at this stage is not affected by the nearby machines or facilities to avoid any structural weakness of the pumping unit and supporting foundation.



## 5. OPERATIONAL MODAL ANALYSIS

Operational modal analysis has a significant advantage over classical modal analysis. Higher estimation accuracy can be obtained by letting the better, spatially distributed, natural load exciting the structure. classical modal analysis using hammer, shaker or multiple shaker excitation, and operational modal analysis based on output-only measurements [9]. For a good simulation, the pumping system is represented by 36 node, 56 line, and 34 surfaces as shown in Fig.9. The measurements were done at speed of 30 Hz (60% of rating speed) because it is the most serious speed of the system that can cause resonance problem.

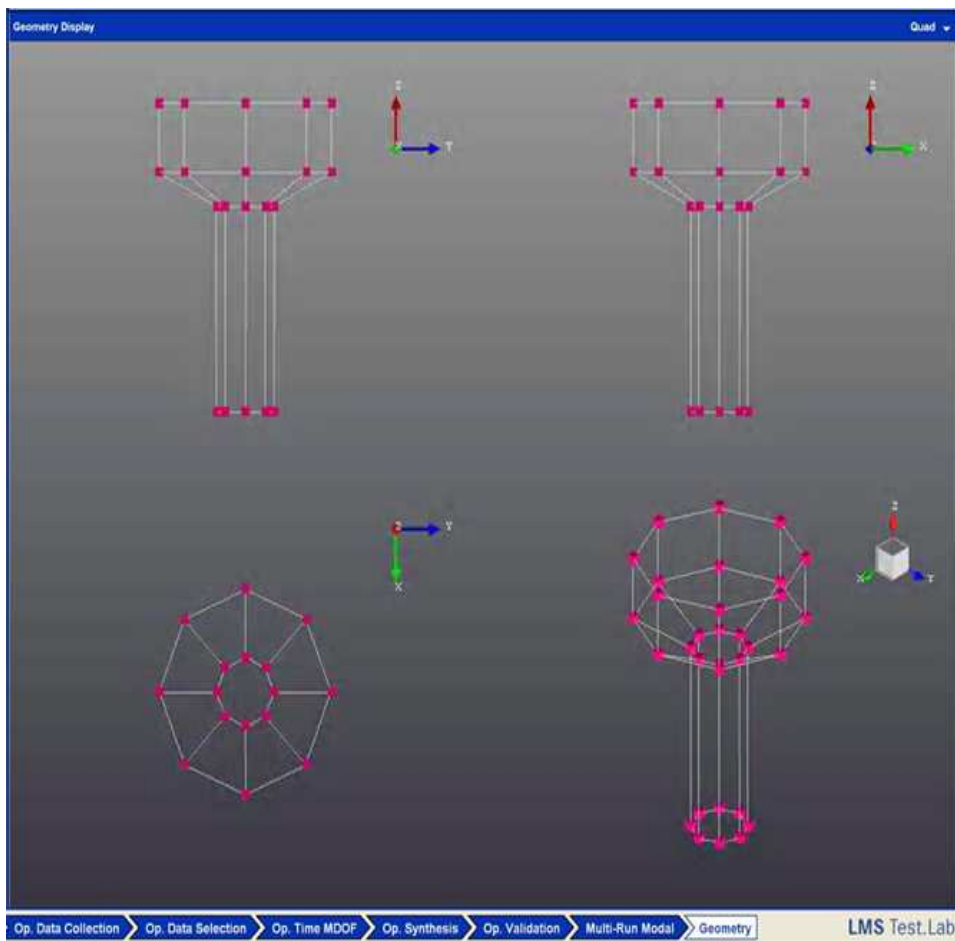


Fig. 9 Pump unit represented as nodes, lines, and surfaces

Table 2 shows the first 7 modes obtained from 8 to 1000 Hz. According to these results it is indicated that the most serious mode shape is at 30.62 Hz (66 %) so, it is very important to modify this mode to avoid the problem of resonance. The measured frequency response function FRF is shown in Fig. 10.

**Table 2 Modal parameters measured from operational modal analysis**

Mode No	Frequency [Hz]	Damping Ratio (%)
1	3062.	0.405
2	86.47	0.7202
3	89.32	0.3155
4	547.1	0.5435
5	562.8	0.1471
6	879.8	0.4224
7	1070	0.08212

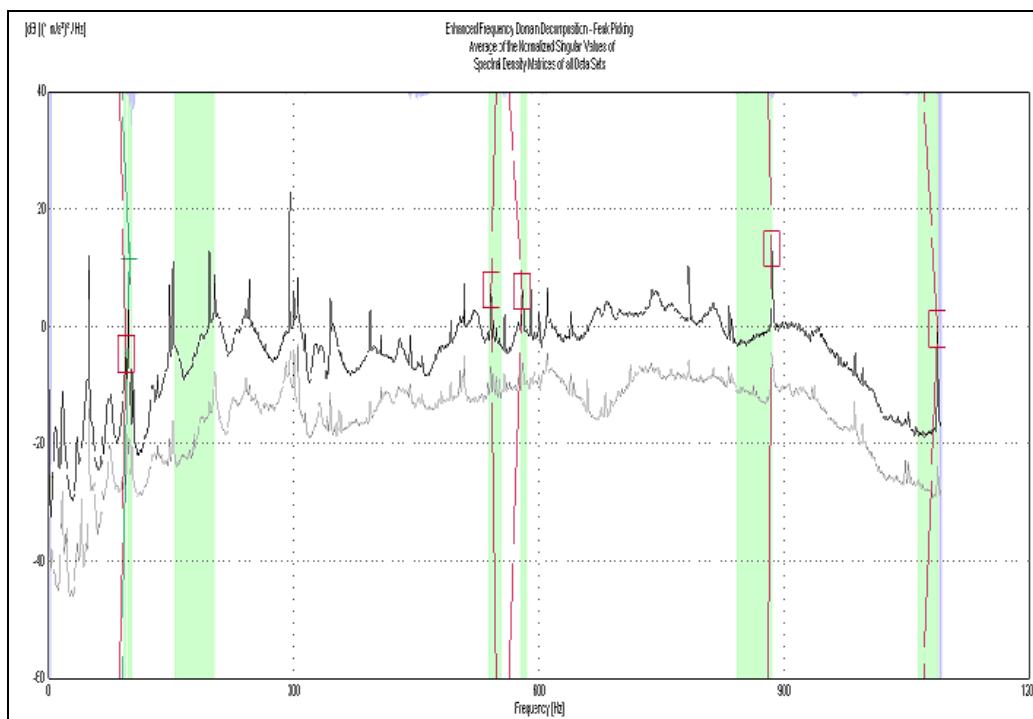


Fig. 10 Frequency response function of the pumping system at 60% of rating speed

Figure 11 indicates that the natural modes are extremely different from the natural modes obtained from case of the pump free supported in air and the case of pump grounded in water because of changing of the boundary condition (the pump is operating at speed 30 Hz) except at the resonance frequency. The presence of water reduces the natural frequencies and enhances damping. Some modes disappeared and other shifted to other values due to effect of water filling the pumping system. Water

worked as a natural dashpot which affects the modes and increase damping. Modal tests at actual operation conditions help to solve existing structural problems and modify any future dynamic behavior based on actual operational and fundamental modes.

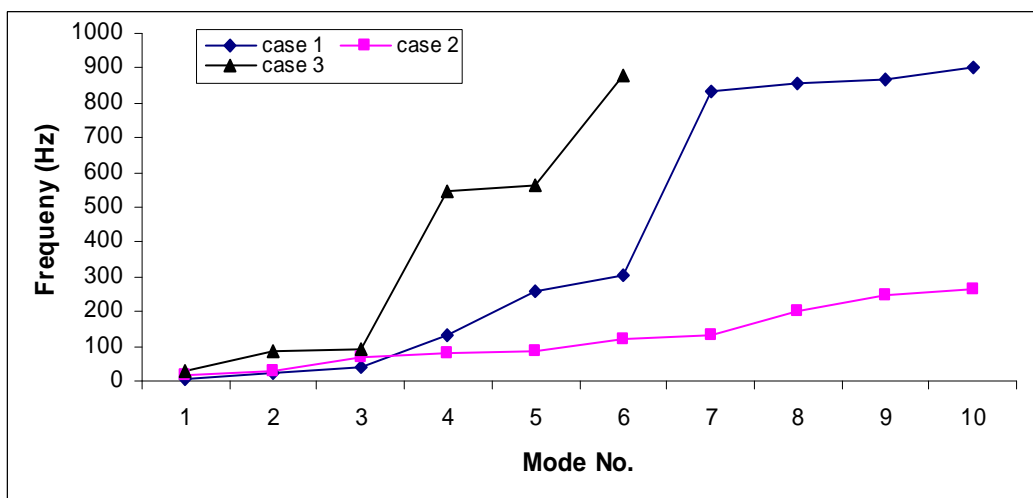


Fig. 11 Measured modal frequencies change for different three cases

### 6. FINITE ELEMENT ANALYSIS

A 3D finite element (FE) model of the pump unit was built to predict the natural frequencies and mode shapes for the pump unit free support in air, and for the pump is immersed support in water using ANSYS software is shown in Fig.12, 13. The proposed model geometry is in exact dimensions used in the experimental setup. The proposed material models used for the pump model are defined as an isotropic material with young’s modulus and poison’s ratio value and all of them acts as structural, linear, and elastic materials as shown in Table 3. A free mesh has been used with the element type of SOLID 95. Block lanczos eigenvalue solver is used for solving the model. Results of experimental and analytical modal analysis are shown in Table 4.

Table 3 Material properties

properties	Cast iron	Stainless steel	antimony
Young’s modulus, (GPa)	2.1 E+011	2.1 E+011	6.9 E+010
Poisson ratio	0.3	0.3	0.3
Density (kg/m <sup>3</sup> )	7860	8100	2710

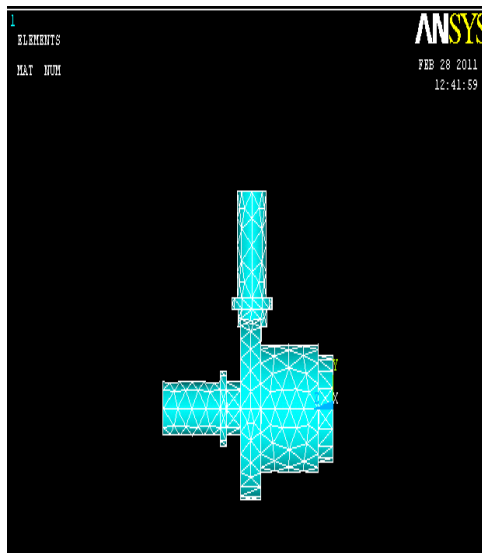
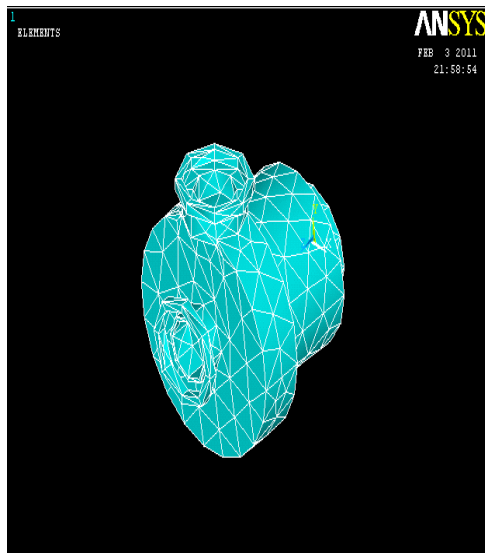


Fig.12 FE model of the pump unit free support in air Fig.13 FE model of the pump unit grounded support in water

**Table 4 Correlation between analytical and experimental modal frequencies**

Mode	Pump unit free support in air		Pump unit grounded support in Water	
	analytical	experimental	analytical	experimental
1	5.0665	4.9	15.69	15.24
2	24.405	23.80	31.802	30.0
3	38.729	37.70	70.165	68.40
4	133.13	130.12	84.241	81.20
5	267.04	260.60	87.505	86.0
6	314.81	307.22	125.05	122.40
7	854.09	833.51	134.75	130.0
8	886.95	865.57	207.54	200.0
9	870.39	858.54	251.37	248.50
10	907.62	882.68	268.545	265.36

The results showed good correlation between both experimental and analytical analyses for the two cases. Deviation in the order of 2.0 % for the modes of the model free support in air, however, deviation of the FE and experimental results for the pump unit model immersed support in water, is in the order of 3.2 %. This minor error is due to nonlinearity of the system, approximation of parameters, and real physical models can not be matched 100% as shown in Fig. 14. The results show a great accordance

between the experimental and analytical models and can be used for further analysis damage detection and structural modification. Both analytical and experimental models were carried out on a pumping system to define a modal model describing the dynamic characteristics of the system. The models are used for understanding of structural characteristics, operating conditions, and performance criteria that enables to design of optimum dynamic behavior or solving structural dynamic problems of the system. The results indicate the importance of doing modal testing in the actual operation and boundary conditions. The dangerous rotating speeds are well determined to avoid operation at resonance condition. The proposed FE model can be used to predict vibration characteristics of variable speed pumping stations under different operating and environmental conditions.

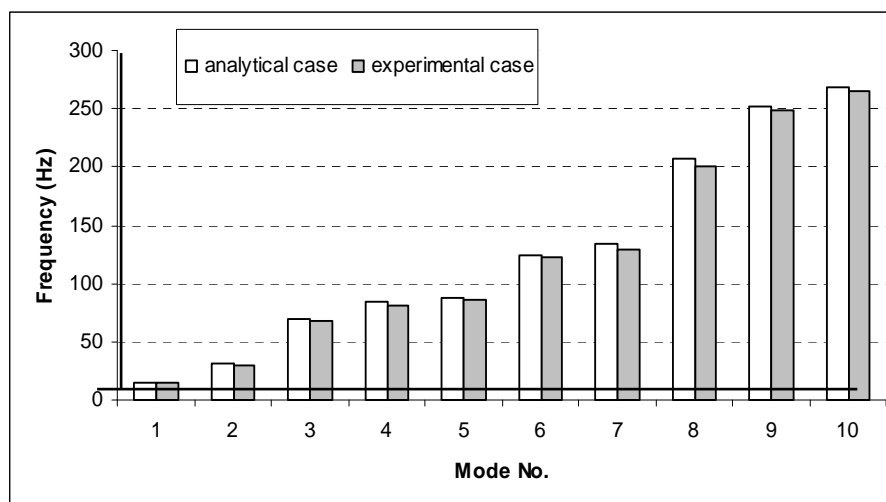


Fig. 14 Correlation between analytical and experimental modal frequencies

## 7. CONCLUSION

- Vibration level increases slightly with increasing the running speed as long as the running speeds are not at resonant frequency.
- It is necessary to do modal analysis at different conditions to predict the dynamic behavior.
- Operating at resonance frequency affects performance and increase vibration level in obviously way.
- The results indicate that resonance is a major problem that can be encountered in variable speed pumps.
- Natural frequencies obtained from experimental modal testing are extremely different from that obtained from operational modal analysis.
- It is important to evaluate the dynamic behavior of variable speed pumps and determine the dangerous operating frequencies to avoid occurring of mechanical problems such as resonance.
- Yet, while variable speed drives clearly offer versatility to the pumping industry and cost savings to the consumer, they also present some difficult engineering challenges for both suppliers and users particularly in the area of vibration.

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### تحليل الاهتزازات لمنظومة الطلبات متغيرة السرعة

إن استخدام المحركات متغيرة السرعة من أفضل الوسائل الاقتصادية لتحسين أداء طلبات رفع المياه لما لها من مميزات بالمقارنة مع الوسائل التقليدية الأخرى. من هنا تظهر مميزات استخدام المحركات متغيرة السرعة في تشغيل الطلبات حيث أنها تؤدي إلى تحسين أداء هذه الطلبات وتسمح لها بأن تعمل عند أقصى كفاءة ممكنة مما يؤدي إلى تقليل الفقد في الطاقة وبالتالي تقليل تكاليف التشغيل. حتى الآن وبالرغم من كل مميزات المحركات متغيرة السرعة إلا أنها تواجه مشاكل كثيرة في مجال الاهتزازات الديناميكية. إن إمكانية تغيير السرعات في هذا النوع من الطلبات يؤدي إلى زيادة فرصة حدوث توافق بين سرعات التشغيل والسرعات الحرجة. فعندما تتغير السرعة تصبح ترددات الأثرية أيضاً متغيرة وبالتالي تزيد من فرص توافق أي من هذه الترددات مع الترددات الطبيعية مما يؤدي إلى حدوث مشكلة الرنين هذا بالمقارنة بالطلبات الثابتة.

لهذا الأسباب فإن هذه الدراسة قد خصصت لعمل مخطط مناسب لفحص وتحليل الأداء الديناميكي للطلبات متغيرة السرعة ومدى حساسية لعملية تغيير السرعات. تم عمل الاختبارات في المعمل على نموذج الدراسة المكون من طلبية طاردة مركزية تعمل بمحرك متغير السرعة بقدرة 5 كيلو وات وعدد لفات 3000 لفة/دقيقة. تم

اختيار مدى من السرعات يتراوح من 25 هيرتز حتى 50 هيرتز. تم قياس الاهتزازة القسرية عند مدى مختلف من السرعات يتراوح من 25 هيرتز حتى 50 هيرتز وذلك لتحديد مستويات الاهتزاز وكذلك ترددات الاثارة. تم تحديد الترددات الخطيرة وذلك لتفادي التشغيل الفعلى عندها. تم عمل التحليل النمطى التجريبي على الطلبة بالمعمل عند مراحل التصميم والتركيب وذلك لتقدير الحالة الديناميكية للمضخة عند الظروف المختلفة. ولقد تم ايضا عمل التحليل النمطى التشغيلى للمضخة وذلك لتحديد الحالة الديناميكية لها عند الظروف الفعلية للتشغيل. وقد اشارت النتائج المعملية ان الخصائص الديناميكية تتغير فى المراحل المختلفة للتصميم والتركيب والتشغيل مما يؤكد ضرورة عمل التحليل النمطى عند الظروف المختلفة لامكانية التحكم فى السلوك الديناميكي ومستوى الاهتزازة. تم بناء نموذج رياضى باستخدام طريقة العناصر المحددة وذلك لتحديد الخصائص الديناميكية للمضخة نظريا وقد تم حل هذا النموذج باستخدام حزمة البرامج التى تعمل على منظومة الحاسب الالى. وقد اظهرت النتائج وجود ارتباط جيد بين النموذج الرياضى والنتائج المعملية.