OUTPUT-ONLY MODAL ANALYSIS OF AN ELECTRICAL SUBMERSIBLE PUMP INSTALLED IN A MOCK WELL USING TIME AND FREQUENCY DOMAIN TECHNIQUES

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Abstract. *Electrical submersible pumps (ESP's) are applied in the petroleum industry to pump large amount of fluids from subsea deep wells. Currently, ESP's are responsible for, approximately, 10% of the world's crude oil production. ESP's are installed inside or near the production well, which means that in offshore ultra-deep water scenarios, maintenance is prohibitive because of the need for expensive and unavailable rig platforms. Assembly errors and manufacturing defects must be avoided to make the method feasible and to prevent premature failure and, this goal can only be achieved through rigorous quality control, operations procedure, qualification and dynamic tests. In this paper, the modal parameters of an ESP installed in a mock well were experimentally identified using the Eigensystem Realization Algorithm (ERA) and Enhanced Frequency Domain Decomposition (EFDD) techniques. In the experimental test, an impact hammer was used to excite the ESP structure and vibration signals were acquired along the ESP structure. Results showed that the impact test is a useful method for modal parameters identification of ESP's, in the operating range. However, due to the location where the impacts were applied, the lower vibration modes, outside the operating range, were not excited.*

Keywords: *Electrical Submersible Pump, Mock Well, Output-Only Modal Analysis, Eigensystem Realization Algorithm, Enhanced Frequency Domain Decomposition.*

1 INTRODUCTION

Electrical submersible pump (ESP) installed in production wells is an efficient and reliable system of artificial elevation capable of bring up great amounts of oil from its natural reservatories, located hundreds of meters down to the sea surface. These volumes vary from a minimum level of 150 barrels per day to a maximum of 150.000 bpd (24 to 24,600 m3/d). However, ESP's varying velocity controllers may yet extend significantly this interval, which implies that these systems has become the most used technology in the world and a natural choice for petroleum reservatories (Takacs, 2009).

Initially, EPS's used to have applicability only in terrestrial production fields (onshore) and, with the pionerism of the Brazilian company, Petrobras, it was initiated successful prototypes tests in maritime fields (offshore) in 1994, in the RJS-221 well, on Campos Basin, being the first one to install the system in deep water, in 1998 (Minette et al., 2016).

This system is installed inside a petroleum well or near it, such that, any maintenance type has a high cost in what time and financial resources are concerned. In this way, interventions are realized only when there is a system failure or in the cases where the maintenance fares can remotely be done. Due to the high investments on this type of equipment, taking into account the conception phases, construction and assembly costs, interventions and ceasing production time, it is crucially important that the system stays operational most of the time. For a better dynamic EPS's properties comprehension, it was performed experimental tests that simulate the real operating conditions of this equipment in a mock well.

The objective of the present work is to experimentally identify the modal parameters of an 62 stage ESP set in terms of its natural frequencies, damping rates and normal modes in the operating frequency band, using a time domain technique known as ERA (Eigensystem Realization Algorithm) and a frequency domain technique known EFDD (Enhanced Frequency Domain Decomposition).

2 OUTPUT-ONLY MODAL ANALYSIS

The great limitation of the Experimental Modal Analysis (EMA) is the necessity of knowing the excitation forces magnitudes acting on the tested system, so that it becomes possible to estimate its modal parameters. The force measurement in large structures can be very difficult to be accomplished, if not impossible. To excite a complex structure implies in a considerable economical cost, which can cause local damages, even in the cases of laboratory tests, where it is hard to reproduce the "real life" excitations in a reliable way.

For these reasons, the Output-Only Modal Analysis was developed in order to bypass the existing limitation of the traditional method and, in this way, to offer the possibility of modal parameters assessment only from dynamic response information. It can be mentioned that this analysis allows to estimate these parameters in cases of unknown excitation forces (in terms of its magnitude and frequency), applied to linear and time-invariant systems.

2.1 Eigensystem Realization Algorithm (ERA)

The modal identification method Eigensystem Realization Algorithm (ERA) is a robust method in the time domain, able to deal with close or repeated vibration modes (Juang and Pappa, 1985).

The essence of the ERA is to apply singular value decomposition (SVD) in the free vibration responses to determine the minimum number of the structure degrees of freedom (number of vibration modes) and to determine the system matrix [A], the entry matrix [B] and the output matrix [C] using the space-state representation. The estimated system matrix may be used in the eigenvalue problem and thus determine the modal parameters. When the dimensions of [A], [B] and [C] are minimal, the minimum realization of the system is achieved. A N-degree-of-freedom system with P excitation forces is governed by the Eq. (1) (He and Fu, 2001):

$$[M]_{N\times N}\{x\}_{N\times 1} + [D]_{N\times N}\{x\}_{N\times 1} + [K]_{N\times N}\{x\}_{N\times 1} = \{F\}_{P\times 1} \tag{1}$$

where $[M]$, $[D]$ and $[K]$ are the mass, damping and stiffness matrices of the system, respectively, and $\{F\}$ is the excitation force vector. Combining the equation with an identity matrix $\{x\} = [I]\{x\}$ and defining the space-state variable $\{X\} = \begin{Bmatrix}\{x\}\\ \{x\}\end{Bmatrix}$, we can form the Eq. (2):

$$\{X\} = \begin{bmatrix} [0] & [I] \\ [M]^{-1}[K] & [M]^{-1}[D] \end{bmatrix}_{2N\times 2N} \{X\}_{2N\times 1} + \begin{bmatrix} [0] \\ [M]^{-1} \end{bmatrix}_{2N\times P} \{F\}_{P\times 1} \tag{2}$$

or:

$$\{X\} = [A]\{X\} + [B]\{F\} \tag{3}$$

If the system has L output points, the output vector $\{Y\}$ is related to the state variable through the output matrix $[C]$:

$$\{Y\}_{L\times 1} = [C]_{L\times 2N}\{X\}_{2N\times 1} \tag{4}$$

From the control theory, the impulse response matrix can be written as a function of $[A]$, $[B]$ and $[C]$ matrices according to the Eq. (5):

$$[\overline{H}(k)] = [C][A]^{k-1}[B] \tag{5}$$

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In practice, the impulse response matrix can be derived by applying the Laplace inverse transform to the transfer functions. By sampling the impulse response function (t), we have a time series for the (k) where $= 1, 2, \ldots$. The impulse response matrix considering all measurements is given by the Eq. (6):

$$[\bar{H}(k)] = \begin{bmatrix} {}_{11}(k) & {}_{1P}(k) \\ {}_{L1}(k) & {}_{LP}(k) \end{bmatrix}_{L \times P} \tag{6}$$

The minimum realization becomes a math task to find $[A]$, $[B]$ and $[C]$ matrices from $[\bar{H}(k)]$, so that the matrix order is minimal. From the solution of the eigenvalue problem, with $[A]$ matrix (called also system matrix), the natural frequencies and damping rates could be calculated (He and Fu, 2001). Assuming that the r-th eigenvalue of the $[A]$ matrix is of the form the Eq. (7):

$$e^{s_r \ t} = a_r + jb_r \tag{7}$$

and knowing that each s_r is of the form given by Eq. (8):

$$s_r = \quad n_r + j\omega_{dr} \tag{8}$$

then, combining equations (7) and (8), we get the Eq. (9) and Eq. (10):

$$n_r = \quad \frac{1}{2\Delta t} \ln(a^2 + b^2) \tag{9}$$

$$\omega_{dr} = \frac{1}{\Delta t} \tan^{-1} \frac{b_r}{a_r} \tag{10}$$

so, the undamped natural frequencies and the damping rate for each vibration mode can be derived by Eq. (11) and Eq. (12):

$$\omega_r = \sqrt{\omega_{dr}{}^2 + n_r{}^2} \tag{11}$$

$$\zeta_r = \frac{n_r}{\omega_r} \tag{12}$$

The matrix of vibration modes $[\Phi]$ is obtained by multiplying the $[C]$ matrix and the eigenvectors matrix $[\Psi]$, from $[A]$, according to the Eq. (13):

$$[\Phi] = [C][\Psi] \tag{13}$$

Each column of $[\varphi]$ represents a modal shape of the system. The confidence factor used in the ERA method to differentiate the real modes from the numeric modes is the Modal Assurance Criterion (MAC) (Juang and Pappa, 1985). The MAC value is used to determine the similarity (linearity) of a mode shape when it is compared with another reference mode. This value varies from 0 to 1, where 0 corresponds to the inconsistent modes, and 1 corresponds to consistent ones. The MAC value is calculated from the Eq. (14) (Caicedo, 2011):

$$MAC_{c,d} = \frac{\left\{\sum_{j=1}^{n} \Phi_{c,j}\Phi_{d,j}\right\}^2}{\left\{\sum_{j=1}^{n}\left(\Phi_{c,j}\right)^2\right\}\left\{\sum_{j=1}^{n}\left(\Phi_{d,j}\right)^2\right\}} \tag{14}$$

where $_{c,j}$ is the j-th coordinate of the mode shape c and $_{d,j}$ is the j-th coordinate of the mode shape d. As the real mode is unknown a priori, MAC is used to compare the identified modes in each iteration from the stabilization diagram for a particular frequency. The result of this operation is a vector indicating which modes are similar to the reference mode.

In the modal identification of a dynamic system in the time domain, the order of the system (vibration modes) is not known a priori. Therefore, it is customary to predefine a model order greater than that one expected in practice. During the calculations, this high order system leads to the appearance of computational or numerical modes mixed to the real modes (Ni et al., 2012).

One of the techniques used to separate the real modes from the computational modes, is through the utilization of the stabilization diagram. It is known that the structural modes remain stable when changing the system order, while the computational ones do not. If the modal parameters do not vary much within a predetermined tolerance range, when the modal order varies then the mode is considered stable and thus considered real. Otherwise, the variation of the modes is outside the tolerance range then the mode is considered spurious or computational.

In this study, the stabilization diagram was used with the natural frequencies values identified in Hz (X axis) as a function of the order number (Y axis). Overlapped to the diagram, the ANPSD curve (*Average Normalized Power Spectrum Density*) was also plotted (Minette et al., 2016). This curve is the average of the normalized power spectrum of the vibration signals used in the modal parameter estimation.

2.2 Enhanced Frequency Domain Decomposition (EFDD)

This modal identification technique is an enhanced version of the Frequency Domain Decomposition (FDD) method (Brincker et al., 2000). Its main advantage, related to the FDD method, is the possibility of a more precise assessment of the natural frequencies and the identification of damping rate.

The essence of the EFDD method is the Power Spectrum Density (PSD) function obtained from the evaluated system response information which graphical representation permits to identify the system natural frequencies and corresponding vibration normal modes.

The estimated PSD matrix decomposition, $\mathrm{G_y(f)}$, formed by the estimated PSD's for all the sensors, may be realized using the Singular Value Decomposition (SVD), which for a complex hermitian and positive definite matrix has the following form:

$$\mathbf{G}_y(f) = \mathbf{USU}^{\mathrm{H}} \tag{15}$$

where the diagonal matrix $\mathbf{S}$ is the singular value matrix and the unity matrix $\mathbf{U} = [\boldsymbol{u}_1, \boldsymbol{u}_2, \dots]$, contains the singular vectors which are interpreted as the modal vectors of the system. It must be mentioned that the SVD method, shown in Eq. (15), does not fully correspond to the theoretical decomposition of the PSD matrix and, hence, the FDD method is an approximate solution. This approximation is studied by two different perspectives; the first one is the PSD decomposition supposing a white noise excitation, and the second one is the PSD decomposition supposing that the modal coordinates are not correlated.

Under random vibrations, it is a commonly assumed approximation that the modal coordinates are not correlated and this supposition may be used as a basis for the modal decomposition of random signals which is approximately valid in cases where the excitation cannot be considered as a white noise, what would happen in the case of impact tests, which, in turn, is the case study of this work and, for this reason, will be described as follows.

In the general damping and complex modes cases, the frequency domain modal decomposition of the PSD matrix, assuming uncorrelated modal coordinates, is given by:

$$\begin{aligned}\mathbf{G}_y(w) = \mathbf{A}[Q_n(\ \ w)Q_n(w)]\mathbf{A}^T + \mathbf{A}[Q_n(\ \ w)Q_n(w)]\mathbf{A}^H + \mathbf{A}\ [Q_n(\ \ w)Q_n(w)]\mathbf{A}^T \\ + \mathbf{A}\ [Q_n(\ \ w)Q_n(w)]\mathbf{A}^H\end{aligned} \tag{16}$$

where $Q_n(w)$ are the frequency domain modal coordinates and $\mathbf{A}$ is the vibration modes matrix. Using an approximation given by:

$$\frac{g(w)}{(iw\ \ \ \lambda_n)}\ \ \ \frac{g(w_{dn})}{(iw\ \ \ \lambda_n)} \tag{17}$$

and taking into account that the diagonal terms of Eq. (16) can be approximated by:

$$Q_n(\ \ w)Q_n(w)\ \ \ \frac{g_{1n}}{(iw\ \ \ \lambda_n)} \tag{18a}$$

$$Q_n(\ \ w)Q_n(w)\ \ \ \frac{g_{2n}}{(\ \ iw\ \ \ \lambda_n)} \tag{18b}$$

$$Q_n(\ \ w)Q_n(w)\ \ \ \frac{g_{3n}}{(iw\ \ \ \lambda_n)} \tag{18c}$$

$$Q_n(\ \ w)Q_n(w)\ \ \ \frac{g_{4n}}{(\ \ iw\ \ \ \lambda_n)} \tag{18d}$$

replacing the approximations of Eq. (18) into Eq. (16) and considering that each element in the matrix can be modally composed, the following spectral density matrix approximation is obtained:

$$\mathbf{G}_y(w) \quad \sum_{n=1}^{N}\left(\frac{(g_{1n}\boldsymbol{a}_{n+}g_{3n}\boldsymbol{a}_n)\boldsymbol{a}_n^T}{iw \quad \lambda_n} + \frac{(g_{2n}\boldsymbol{a}_n + g_{4n}\boldsymbol{a}_n)\boldsymbol{a}_n^H}{iw \quad \lambda_n}\right) \tag{19}$$

For real modes and low damping, the $g_{1n}, g_{2n}, g_{3n}, g_{4n}$ terms are the equal for the same mode, hence, Eq. (16) reduces to Eq. (20) (Brincker and Ventura, 2015) as:

$$\mathbf{G}_y(w) \quad \sum_{n=1}^{N} 2g_n\left(\frac{\boldsymbol{a}_n\boldsymbol{a}_n^T}{iw \quad \lambda_n} + \frac{\boldsymbol{a}_n\boldsymbol{a}_n^T}{iw \quad \lambda_n}\right) \tag{20}$$

In this case, assuming that the system has uncorrelated modal coordinates, Eq. (16) may be decomposed by the SVD method in the form of Eq. (15) (Brincker and Ventura, 2015). In this part, it is necessary to use the user criterion in order to recognize the peaks in the singular value graphical presentation, considering the possibility of finding low dynamic range peaks which may be difficult to identify due to the presence of noise.

With the region of analysis selected using MAC (for each mode of vibration, corresponding to a SDOF system), the corresponding Autocorrelation Function is calculated by the Inverse Fast Fourier Transform (IFFT). Next, a triangular time window is applied in order to enhance the Autocorrelation Function estimative. Finally, through the logarithmic decrement it is possible to calculate the corresponding damping rate and with the Zero Crossing curve it is possible to calculate the undamped natural frequency of the SDOF system.

3 EXPERIMENTAL ANALYSIS

3.1 Test Device

The tested ESP set has a 19.44 m length and 62 stages. Fig. 1 shows the mock well circuit schematic installed in Mossoró/RN, with the ESP's dimensions and the local where the impacts were applied. A more detailed technical description may be found in (Minette et al., 2016).

The impact tests were applied on the production pipe, near the place where the EPS is supported. About 30 impacts (with 3 seconds of duration) were applied on the structure and the vibration signals were originally acquired with a 25 kHz sampling rate (Minette et al., 2016).

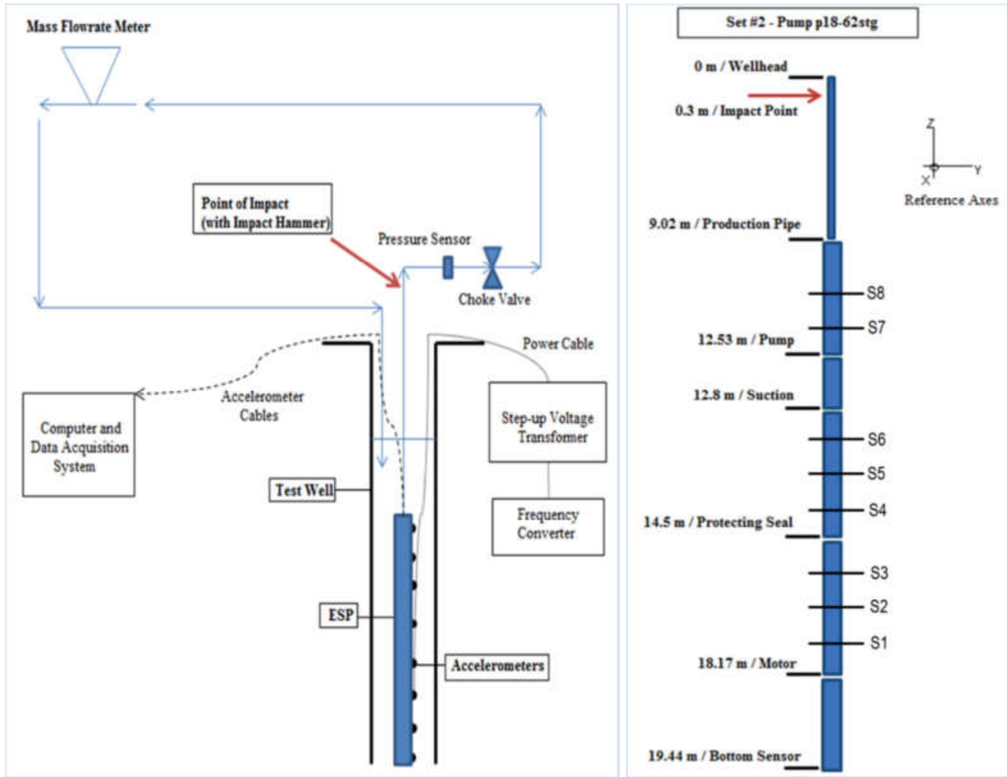

Figure 1. a) Mock well circuit. b) Accelerometers distribution.

3.2 Experimental Results using ERA

Figure 1 shows the point where the impacts were applied and the accelerometers locations used for this analysis, totalizing 8 measuring points (S1, S2, …, S8). To estimate the modal parameters, only the accelerometers whose responses to the impact represented a free vibration signal, without distortion caused by noise, were considered. Vibration signals were filtered with a500 Hz low-pass filter and re-sampled with a 1 kHz sampling rate.

The following stability criteria were defined to mount the stabilization diagram: 1) modes with damping rate greater than 50% were rejected; 2) natural frequencies within the ±1 Hz range were considered good candidates; 3) stabilized modes were defined as those that show a MAC number greater than 60% and finally; 4) if the number of stabilized modes was larger than 30, then that natural frequency was considered as stable.

Figure 2 shows the stabilization diagram for natural frequencies identified through the ERA method, in the range from 0 to 100 Hz. The red line is the ANPSD (Average Normalized Power Spectrum Density), which represents the average of the normalized power spectra of the vibration signals used for identification.

In the stabilization diagram, the identified natural frequencies correspond to peak values of the ANPSD.

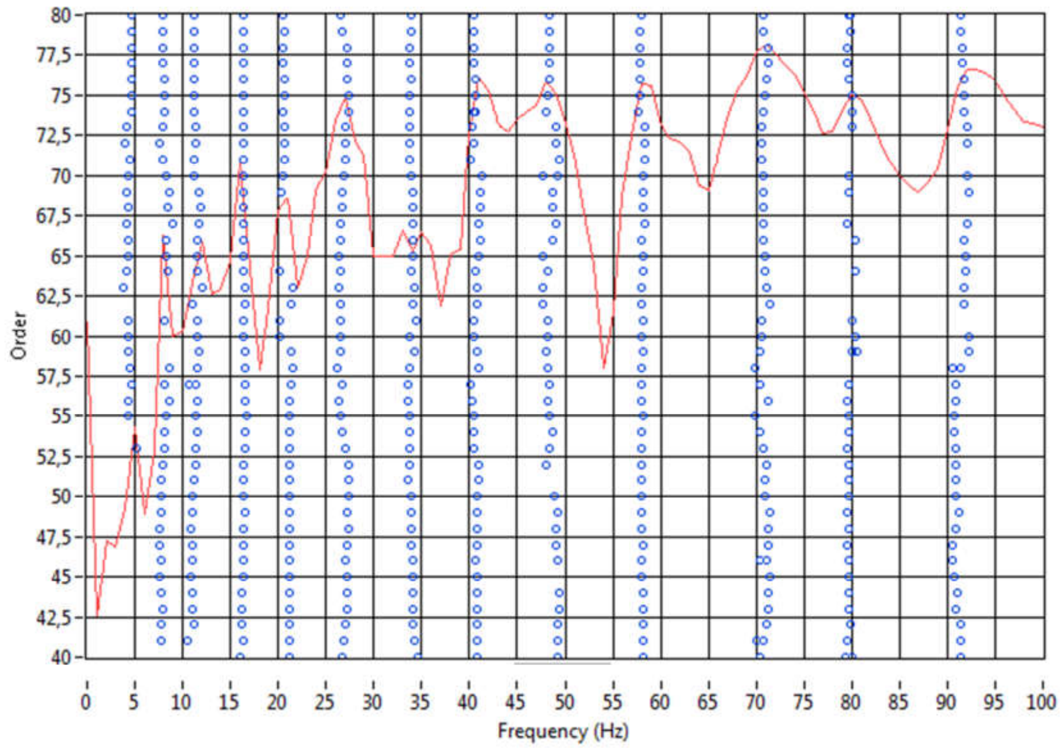

Figure 2. Stabilization diagram for natural frequencies

Table 1 shows the natural frequencies and damping rates that stabilized, in the range from 0 to 100 Hz. These values were obtained when it was considered that the maximum number of modes presented in the free vibration signals was 80.

Table 1. Modal parameters of the ESP identified using ERA

Mode	Frequency [Hz]	Damping Rate [%]
1	4,68	4,48
2	7,98	2,27
3	11,27	3,32
4	16,32	2,13
5	20,56	3,05
6	27,96	5,12
7	33,88	5,06
8	40,53	1,12
9	48,37	2,06

10	57,96	2,51
11	70,74	3,02
12	79,85	4,56
13	91,34	2,58

3.3 Experimental Results using EFDD

It was used the same combination of sensors of the previous analysis, which means that only the 8 ones in the Y direction were considered, which is the same direction of the impacts (see Fig. 1). The vibration response signals were filtered and had its mean removed, being re-sampled with a 250 Hz sampling rate, once the interest bandwidth went up to 100 Hz. The singular values graphical output (Fig. 3) was built from the estimated spectral density matrix.

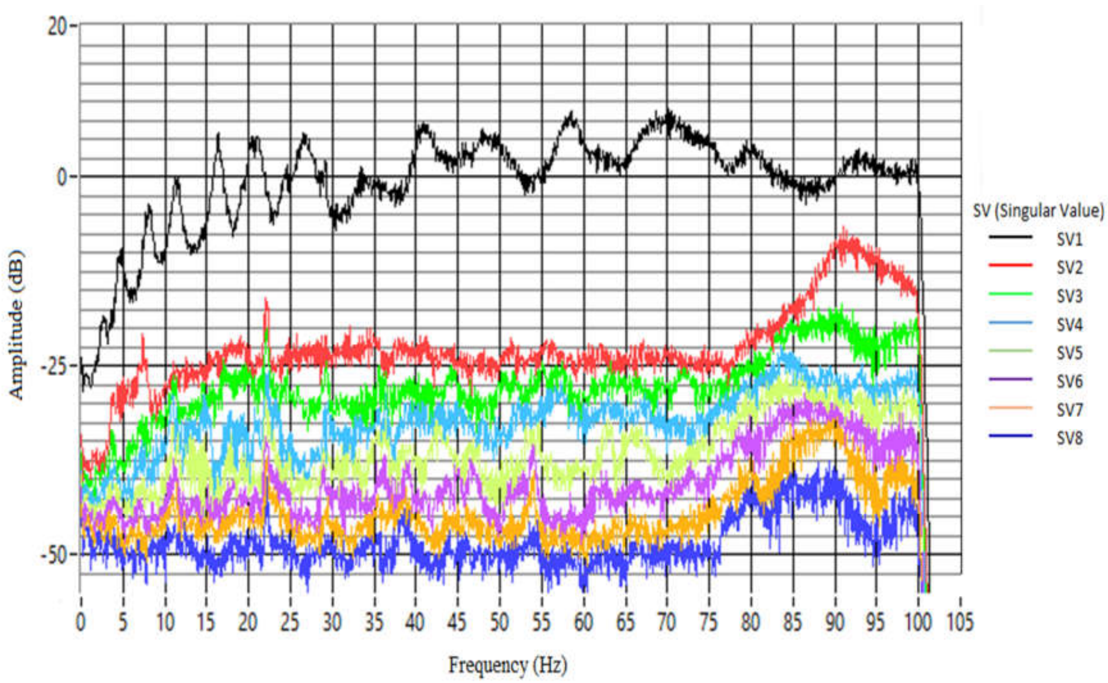

Figure 3. Graphical output of the 8 accelerometers single values parallel to the impacts

Table 2 shows the results obtained with the EFDD technique. From this table it is verified that the modal parameters corresponding to the 7th and the 13th vibration modes (33.88 Hz and 91.34 Hz, respectively) were not identified with the EFDD technique. The motive for this becomes clear when the singular values near these frequencies are analyzed, as shown in Fig. 3, with these peaks having low energy and high noise. A solution to smooth the singular values curve is to minimize the noise using other PSD's estimators, for example, as suggested by some researchers (Brincker and Ventura, 2015).

Table 2. Modal parameters of the ESP identified using EFDD

Mode	Frequency [Hz]	Damping Rate [%]
1	4,73	4,66
2	7,95	2,79
3	11,46	3,42
4	16,48	2,15
5	20,96	3,17
6	26,88	5,24
7	-	-
8	41,04	1,43
9	48,55	3,29
10	58,01	2,49
11	70,51	3,13
12	79,33	4,86
13	-	-

3.4 Results Discussion

Table 3 shows the results obtained with the two techniques (ERA and EFDD). The differences between the ERA and the EFDD, in the estimation of natural frequencies were lower than 2%. Only the difference for the 6th mode was about 4%, showing a good agreement between the results.

Table 3. Differences (%) between modal parameters identified with EFDD and ERA.

Mode	Differences (%) between EFDD and ERA techniques	
	Frequency	Damping
1	1,03	4,06
2	0,42	22,88
3	1,66	2,98
4	1,00	0,98
5	1,93	4,01
6	3,86	2,40
7	-	-

8	1,26	28,03
9	0,38	59,62
10	0,09	0,96
11	0,32	3,63
12	0,65	6,64
13	-	-

4 CONCLUSIONS

In this paper, an ESP set of 62 stages installed in a mock well had its modal parameters experimentally identified using only the structure's vibration responses obtained through impact tests. Two techniques were used: ERA (time domain) and the EFDD (frequency domain) methods.

It is concluded that the impact test is a valuable method for identifying the ESP modal parameters in a mock well within the operating range. However, if the rotordynamic effect is to be considered, the use of Operational Modal Analysis (OMA), where the equipment is tested during normal operation, should be considered. This will be the next step of this research.

The damping rate is the most challenging parameter to estimate, as can be seen in Table 3, with differences reaching 59,62%. This problem is independent of the estimation method and still is object of actual researches (Rainieri and Fabbrocino, 2014). Hence, the use of the estimated damping rates must be cautious. On the other hand, the natural frequencies identification does not represent the same challenge, as shown along this work.

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