Condition monitoring and diagnostic of an extruder motor and its gearbox vibration problem

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Condition monitoring and diagnostic of an extruder motor and its gearbox vibration problem

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Abstract Reliable production plants are vital in today's economy as they have a direct impact on productivity, profitability and overall prosperity of industrial nations whose income depends on their industries, products they create and exports. Reliability depends on regular condition monitoring and plant maintenance and more importantly the tools and technologies being designed, developed and used. Excess vibration is one source of plant failure and this can have many root causes ranging from dynamic incompatibility between different elements of machinery or their foundations to other causes resulting from general wear and tear. Prediction or detection of source or causes of these unwanted problems is challenging and requires sophisticated tools and theories as well as experience and expertise. Moreover, classical vibration monitoring on its own cannot predict the root causes of failures that are due to operating conditions, especially when severe or abnormal service conditions are present. Under such conditions, the systems may behave in a completely different or unpredictable manner: active excitation forces/loads and resulting displacements can make the system behave in a nonlinear fashion. Here an industrial case study that involves the non-destructive evaluation of an extruder motor due to excessive vibration is presented. A novel Dynamic Design Verification (DDV) procedure for non-destructive monitoring and resonant vibration identification - which relies on a combined Experimental Modal Analysis (EMA), Operating Deflection Shapes (ODS) and linear elastic Finite Element Analysis (FEA) - is used to assess the structural integrity and dynamic behaviour of extruder motor and associated subsystems. The analyses concluded that the root cause of the high vibration is not due to wear and tear of the motors but is due to weakness of the motor support structures and concrete foundation supporting the extruder motor. Based on the performed analyses, structural dynamic modifications (additional supports) applied to the 'non-drive end' of the motor have been considered and their effects on the system are analysed. It was concluded that this modification shifted the lowest natural frequency away from the operating speed and effectively reduced vibration to a safe level.

Keywords Rotating machinery, vibrations, modal analysis, operating deflection shape analysis, non-destructive testing, dynamic design verification

1. Introduction

Detection and monitoring of vibration in rotating machines is an important and effective way to identify possible failures due to poor design and maintenance, imbalance problems and/or incompatibility between the supporting structure/foundation, machine parts and other auxiliary equipment [1]. However, to understand and pinpoint the root cause of excessive vibration [2] it is often necessary to determine the interaction amongst all the mentioned components which may be very complex. A review of the progress made in rotating machinery condition monitoring and diagnosis through the use of different signal processing techniques for vibration analysis is considered in [3]. Different methods including automated diagnosis were used effectively to obtain vital information and detect machine faults from vibration profile before catastrophic failures (of parts or components).

A parameter estimation technique has been considered in [4] for monitoring machinery condition based on experimentally measured vibration data, as a way for observing or predicting an incipient failure of its components indicated by a change in stiffness or mass element. An intelligent platform based on a standalone data-driven approach is proposed in [5] to diagnose faults such as performance degradation assessment, remaining life estimation and dynamic design verifications of steady-state in-service rotating machinery. To show the effectiveness of the developed intelligent platform in different applications two industrial case-studies were also considered in [5].

As mentioned in [6] excessive vibrational amplitudes - resulting in deflection of the machine parts and large displacements - are most influenced by the behaviour of the foundation, which changes the overall dynamic response of the system, making it behave in a completely different and unpredictable manner. A new procedure to identify the modal parameters of a rigid foundation – which affects the vibration behaviour of a rotor supported by two hydrodynamic bearings - was developed in [7]. A component synthesis method for vibration isolation/damping of massive rotating machinery supported by an elastic structure is considered in [8]. The method allows easy calculation of the stiffness matrix of resilient elements as well as the system vibration characteristics.

The dynamical behaviour of a rotating machines placed on flexible foundation structure is studied in [9] using the mechanical impedance method. The technique incorporates both theory and experiments for the system modelling, and uses the experimental results for validation. A mathematical methodology - using modal parameters (obtained experimentally through frequency response functions and Fourier Transforms) to analyse the foundation influence on rotating machinery (the rotor-bearings system) and finite element method to model the rotor and reduce the degree of freedom order of the foundation model - was considered in [6,10]. A built-in masses system – similar to a base isolation system - has been considered in [11] as an alternative method to reduce system vibration at points far from the exciting machine. Significant reduction in vibration levels and excess movement has been obtained when a relatively small percentage of the total mass of the machine has been used as additional weight in the supporting structure.

Another way of analysing faults or excess movement in rotating machinery can be obtained through the use of the Operating Deflection Shape (ODS) analysis, a non-destructive and non-invasive technique used to monitor the behaviour of a system while in operation [12-17]. As shown in [17] the ODS technique is especially useful when classical condition monitoring is not possible. Operating Deflection Shapes importance as a means of detecting excessive vibrations due to, faults, unbalance or misalignment in rotating machinery and its components has been highlighted by the studies performed in [18-22]. Significant changes in the ODS data acquired from multiple accelerometers through the frequency spectra of the displacement and acceleration responses [18-20] have been considered as an early warning indicator of rotating machinery faults.

In this paper the outcome of an industrial case study for the vibration assessment and analysis of an extruder motor due to excessive vibration is presented. The investigation is based on the combined application of linear elastic Finite Element Analysis (FEA), Experimental Modal Analysis (EMA) and ODS techniques assess the condition of an in-service pipe extruder motor that was showing signs of excessive vibration.

The root cause of these excess vibrations, as discussed above, was the weakness of the motor support and concrete foundation supporting the extruder motor. The analysis also reveals that the extruder motor and gearbox has been dynamically well designed with its natural frequencies located well above its intended operating speed. However, from the EMA and FEA simulation, the motor support and concrete foundation does not achieve its designed stiffness causing this “stiffness drop” to shift the foundation natural frequencies into the extruder operating region.

To address the problem, additional supports have been added to the “non-drive end” of the motor. It was concluded that the performed structural dynamic modification on the system had managed to shift the lowest natural frequency away from the operating speed and effectively reduced vibration to a safe level.

2. Problem Formulation and Procedure Description

The need for a non-invasive testing and evaluation approach was justified by the excessive vibrations of a piece of industrial equipment, i.e. an extruded motor, which created an unpleasant working condition for the hired personnel in a nearby control room. Routine vibration monitoring reported/pinpointed high vibration in the horizontal direction of an extruder motor, having the vibration level recorded as high as 5 to 6 mm/sec^2. In this case, the motor shaft and gear is linked with clutch-type coupling. The motor bearing is of sleeve-type whilst the gearbox uses a rolling element bearing.

To solve the excessive vibration problem, a design verification approach [23], which combines Experimental Modal Analysis (EMA) Operating Deflection Shapes (ODS) and Finite Element method (FEM), is considered. The approach adopted was to investigate whether or not any vibration could induce excessive cyclic stresses on both the motor itself, its supporting components and the foundation of the system, thus, affecting the overall structural integrity which may result in a catastrophic failure.

The measurement and evaluation procedure have been devised using a state-of-the-art MDT-Q2 data acquisition system (Quadrant & Technologies SDN. BHD.) based on a 4-channel real-time machinery analyser, tri-axial (measurement locations were taken in the principal directions) and uni-axis accelerometers, modally tuned impact hammer and ME’scope software (Vibrant Technology, Inc., USA) used to analyse the motion and the excessive vibration levels of the motor and its gearbox.

Modal analysis using Frequency Response Function (FRF) measurement technique, (i.e., impact testing) was used in determining the dynamic characteristics namely the natural frequencies, mode shapes and damping. The measured input is force from the impact hammer and the measured output is acceleration from the accelerometer. The test was carried out by fixing the impact hammer and roving the tri-axial accelerometer throughout the measurement points. The sampling rate used was 2048 samples/sec with block size of 4096. This yields frequency resolutions of 0.5 Hz and 2 seconds of time record length to capture every response signal. 5 averages or impacts were taken at each measurement point. The signals were averaged and processed to generate the Frequency Response Functions (FRFs) estimation. The modal extraction technique was performed using modal analysis software called ME’scope. The software was also used to perform post-processing of the acquired data and curve-fitting for the extraction of modal frequency, modal damping and modal shape. In addition to that, a three-dimensional structural model that represents the motor and its gearbox was created. It consists of a series of points with Cartesian coordinate connected together using straight lines to form surfaces. The displayed point numbers show the measurement locations as in the actual motor and its gearbox. This model was used to display the mode shapes of the structures from the acquired data.

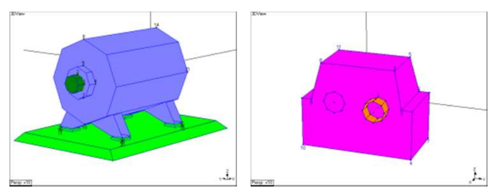
Meanwhile, ODS requires the system to be in steady-state operating condition. The test was performed in similar manner where the impact hammer was replaced with a uni-axis accelerometer as a reference input signal indicating the reference DOF and for the purpose of measuring the relative phase between accelerometers. The tri-axial accelerometer is used as a roving accelerometer and can measure acceleration in three orthogonal directions simultaneously. The roving accelerometer approach allows large structure with multiple points to be investigated cheaply and effectively by passing the need for using many accelerometers simultaneously. The sampling rate used was 2048 samples/sec with block size of 8192. This yields better frequency resolutions of 0.25 Hz and 4 seconds of time record length to capture every response signal. 10 averages were taken at each measurement point. The vibration signal was processed into frequency domain, (i.e., ODS FRF) and being sent to ME'scope for post-processing to generate the deflection shape during operation.

Lastly, Finite Element (FE) Modal Analysis and Structural Design Modification (SDM) using FEA was performed on the motor and its gearbox for verification purpose of the FE model prior to fabrication. Finite element model was created with references to structural drawings and general assembly and layout drawings. The main advantage of SDM in Finite Element Analysis (FEA) allows large number of design modifications without having the unnecessary physical cycles of 'modify-and-test'.

2.1. Modal Analysis (MA)

Modal analysis is used in investigating the dynamic behaviour of static systems. However, Modal analysis enables an enhanced understanding and identification of the root cause of vibration phenomena encountered in engineering by describing a system with its modal parameters - namely the natural frequencies, natural damping and natural modes. These three parameters comprehensively define the dynamic characteristics of a system. Currently, there are two such techniques used to extract these modal parameters, one is classical Experimental Modal Analysis (EMA) [24-26], the other is FEA. Operational Modal Analysis (OMA) [27,28] and the Impact-Synchronous Modal Analysis (ISMA) [29-31], are relatively new techniques and now being widely used in determining modal parameters of in-service machinery, troubleshooting, Structural Dynamic Modification (SDM), force determination, analytical model updating, optimal dynamic design, passive and active vibration control, as well as vibration-based structural health monitoring in aerospace, mechanical and civil engineering [32-36]. Conventional Experimental Modal Analysis (EMA) has limitations in that it requires the system to be in a complete shutdown state; which means no unaccounted excitation forces are induced into the system. OMA holds advantage over EMA in terms of its practicality and simplicity to carry out the procedure and performing the analysis while the system is in operation. However, the lack of knowledge of the input forces does affect the operational modal parameters extracted. Also mode shapes obtained from OMA cannot be normalised accurately, subsequently affecting the development of mathematical models thereafter. However, ISMA [31] has the advantages of the OMA and EMA combined. It carries out the analysis while the system is in operation and is also able to provide the actual input forces in the transfer functions, hence allowing for better/more detailed modal extractions and mathematical model development.

Considering the Dynamic Design Verification (DDV) procedure, modal analysis technique based on Frequency Response Functions (FRF) was used to determine the dynamic characteristics (mode shapes and associated natural frequencies) of the stationary extruder motor and its gearbox [17,37], shown in Fig. 1. Carrying out measurement using Fast Fourier Transformation (FFT) analysis on a continuous system, the FRF, being an estimated quantity, is obtained over a number of averages. The extruder motor (Fig. 1a) has a sleeve-type motor bearing, the motor shaft and gear are linked with clutch-type coupling, and the gearbox (Fig. 1b) uses a rolling element bearing.



(a) (b)

Figure 1. (a) Extruder motor, and (b) Gearbox

The mathematical model of a time varying rotating structure can be expressed as

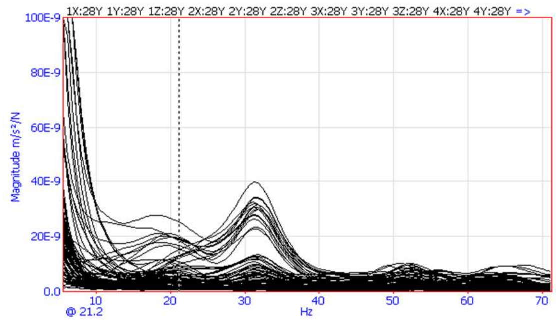
$$[M]\{\ddot{y}\}+[D]\{\dot{y}\}+[K]\{y\}=\{F\} \quad (1)$$

The FRF of the of a viscously damped system of form (1) subjected to a harmonic excitation of the form $\{F(t)\}=\{f(j\omega)\}e^{j\omega t}$ is expressed as

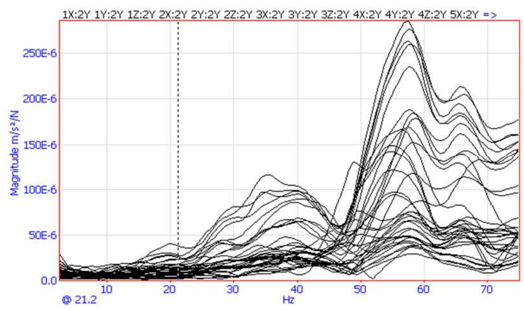
$$[\alpha(j\omega)]=\left[K+j\omega D-\omega^2 M\right]^{-1} \quad (2)$$

Modal analysis was performed on the extruder motor and its gearbox using a real-time data acquisition system, impact hammer, tri-axial accelerometers and ME'scope software. All the measurement locations taken using tri-axial accelerometer have been linked together to obtain the ME'scope wire-mesh model of the extruder motor and the gearbox as shown in Fig. 1.

In Fig. 2, the modal frequencies around the running speed are shown for both the extruded motor and the gearbox. It is quite clear that the modes of interest are the ones close to the running speed of 21.2 Hz / 1272 rpm.



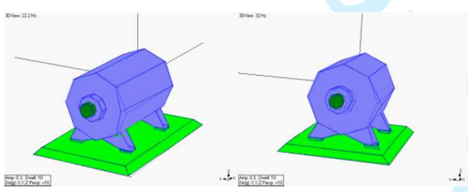
(a)



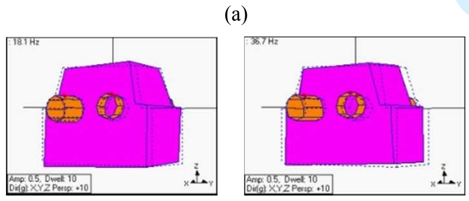
(b)

Figure 2. Modal Frequencies of the (a) Extruder motor, (b) Gearbox

The first two mode shapes of the motor at the natural frequencies of 22.2 Hz and 32.0 Hz, and first two mode shapes of the gearbox at the natural frequencies of 18.1 Hz and 36.7 Hz respectively, are shown in Fig.3a and Fig. 3b.



(a)



(b)

Figure 3. Mode Shapes of the (a) Extruder motor, (b) Gearbox

Listed in Table 1 and Table 2 are the first four modes extracted from the FRF spectrums. The modes of concern - based on the FRF - are identified near the running speed of 21.2 Hz / 1272 rpm of the motor. The 1st natural frequency of both extruder motor and gearbox recorded at 22.2 Hz and 18.1 Hz are the closest to the running speed of 21.2 Hz.

Table 1. Frequency modes for the Extruded Motor

Shape	Frequency(Hz)	Damping(%)
1	22.2	16.1
2	32.0	8.45
3	52.1	4.32
4	63.7	4.59

Table 2. Frequency modes for Gearbox

Shape	Frequency(Hz)	Damping(%)
1	18.1	12.1
2	36.7	7.70
3	55.1	9.23
4	64.2	1.20

2.2. Operating Deflection Shapes (ODS)

Operating Deflection Shapes (ODS) can be defined as any forced motion of two or more Degrees of Freedom (DOFs) - points & directions - on an in-service machine or structure [38]. ODS can be divided into two categories: time domain ODS and frequency domain ODS. Time domain ODS is extremely useful in giving an overall ODS, which can be planar, orbital or 3D for a non-stationary signal such as a transient signal compared to frequency domain ODS. It is recommended to perform frequency domain ODS for stationary signal such as a steady state signal under a constant operating conditions [39].

In the Dynamic Design Verification (DDV) procedure, frequency domain ODS is considered next to determine the extruder motor and gearbox deflection while in operation. The vibrations response $Y(\omega)$ which defines the structure deflection at a particular frequency [15,40] (i.e. the ODS of the system subject to a harmonic excitation) can be expressed by

$$Y(\omega) = \sum_{r=1}^{n} \frac{\{\psi\}_r^T \{F\}\{\psi\}_r}{\omega^2_{r-\omega^2}} \qquad (3)$$

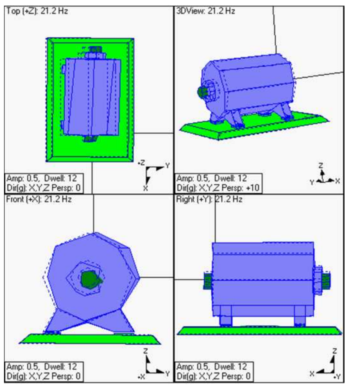
where ω is the excitation frequency, t is the time, F is the harmonic force defined by $\{F\}=\{f\}e^{i\omega t}$: ψ_r is the eigenvector and ω_r is the eigenvalue of the mode r. The local amplitude $Y(\omega)$ in Eq. (1) was obtained as a forced vibration [25] from

$$[M][\psi]\{\ddot{\vartheta}\}+[D][\psi]\{\dot{\vartheta}\}+[K][\psi]\{\vartheta\}=\{F\} \qquad (4)$$

where $[M]$ is the mass matrix, $[D]$ is the damping matrix expressed as a linear combination of $[M]$ and $[K]$, F is the external force, $[\psi]\{\vartheta\}$ represents the forced response and ψ the mode shape vectors.

To capture deflection while in operation, the ODS analysis was performed on the extruder motor and its support using one uni-axial accelerometer as reference (at a reference point) and a tri-axial accelerometer to measure vibration signal (at all the pre-defined measurement locations). Various frequency domain measurements including Fast Fourier Transform (FFT) spectrum, Frequency Response Functions (FRF), or ODS FRF obtained from Cross and Auto Power spectrum can be used to obtain ODS.

(a)



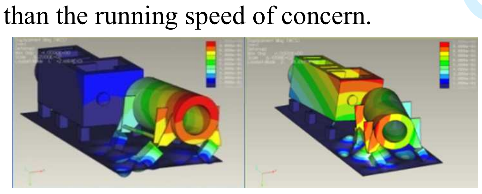
(b)

Figure 4. ODS analysis of the Extruder Motor (a) FFT Spectrum, and (b) Maximum deflection @ Running Speed of 21.2Hz/1272 rpm

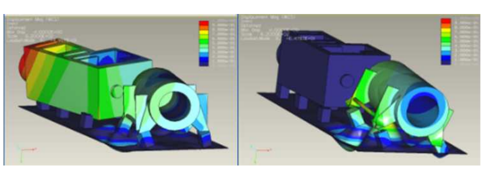
The ODS shapes of the extruder motor obtained using FFT linear spectrum are shown in Fig. 4. It was observed that the FFT spectrum (Fig. 4.a) was dominated by the running speed of the motor at 21.2Hz (1272 rpm). The operating shape at 21.2Hz showed maximum deflection occurs at the motor 'non-drive end' (Fig. 4.b). Moreover, a top view of the ODS shows the extruded motor to be pivoted at the gearbox in-board bearing.

4. Problem Solution

A Finite Element Analysis (FEA) was performed on the extruder motor and its gearbox. The first four modes of vibration generated from the FEA results (Fig. 5) are at 21.5 Hz, 36.6 Hz, 49.7 Hz and 64.8 Hz. It can be observed that 1st natural frequency of both the extruder motor and gearbox from FEA obtained at 21.6Hz are very close to the running speed of 21.2Hz, while the rest of the frequencies were much greater than the running speed of concern.



(a)



(b)

Figure 5. Extruder motor and Gearbox vibration modes (a) 1st mode, (b) 2nd mode, (c) 3rd mode, (d) 4th mode

3.1. Structural Modifications

With reference to the extruder motor and gearbox system design, the Finite Element Analysis (FEA) performed on the 3D model was correlated with the modal analysis as an important part of the verification process [41]. In this case, the correlation established between the modal analysis and Operating Deflection Shape (ODS) analysis results (on one side) and the Finite Element (FE) modal analysis (on the other) indicate a resonance problem due to inadequate supports and weakness of the motor foundation. The ODS study showed extreme vibrations close to the running speed forcing the extruded motor into pivoting/rotating during operation and applying stress at the concrete base foundations (Fig. 6).

(a)

(b)

Figure 6. Concrete base of the (a) extruder motor and, (b) Gearbox

When high levels of vibration propagate from the vibration source, foundation isolation is highly recommended [11]. The critical effect of resonance as a dynamic amplifier was observed when frequencies matched the supporting foundation structure. Due to the mentioned amplification effect, a detailed frequency analysis is often needed to evaluate the assembly's general behaviour.

The supporting element and concrete foundation also formed a an integral part of the overall system and OD showed excessive displacement at the none drive end due to inadequate stiffness of the supporting foundation.

Therefore, the ODS results and the FEA simulation indicate that the motor support and concrete foundation does not achieve the designed stiffness and causes a "stiffness drop". This has subsequently caused a shift in the foundation natural frequencies, bringing it into the extruder motor operating range. Once the weak element of the structure has been identified, Structural Dynamic Modification (SDM) was applied iteratively using ANSYS FEA in order to adjust the foundation stiffness. The adjustment in Fig. 7, applied to the 'non-drive end' of the motor by introducing additional support, aims to stiffen the connection between the motor and the foundation. The modification requires the additional C-Channel to be rigidly fixed to the foundation by anchor bolts, plates and grouting as base (see Fig. 6 and Fig. 7 for details).

Figure 7. Structural dynamic modifications applied to the extruder motor - showing the dimensions and position of the additional leg for the motor at non-drive end

Figure 8 shows the Finite Element Analysis of the proposed correction/adjustment structural modification. The performed structural modification shifted the 1st natural frequency about 40% away from the operating speed while the 2nd natural frequency had remained steady and became the lowest natural frequency for the extruder motor. The resonances of the motor are now well away from the running speed, no pivoting/rotating of the supports appears during operation and the applied stress at the foundation is drastically reduced.

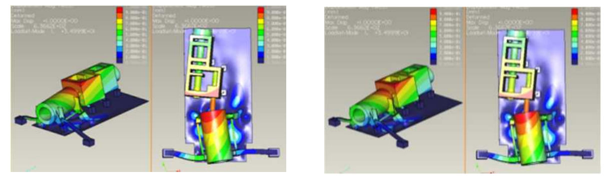


Figure 8. Extruder motor and Gearbox new modes of vibrations (Left) 1st bending mode and (Right) 2nd bending mode

5. Conclusion

In this paper, an investigation into the application of a non-destructive evaluation technique of an extruder motor has been presented. The experimental technique, based on Experimental Modal Analysis (EMA) and Operating Deflection Shape (ODS), was applied for an in-service monitoring of extruder motor vibrations. The ODS study showed that extreme vibrations of the extruder motor was due to its structural natural frequency being "close" to the running speed of the extruder motor which causes "near-resonance" excitation during operation. It was observed that the root cause of the high vibration was due to weakness of the motor support and concrete foundation supporting the extruder motor.

Using a Dynamic Design Verification (DDV) procedure the 1st natural frequency was shifted about 40% away from the operating speed preventing the supports from pivoting during operation and therefore minimizing the applied stress at the foundation. It was concluded that a general requirement for foundation or base natural frequency is that it should be at least 20% - 25% away from the machinery operating speed.

REFERENCES

[1] E. Krämer, 1993. Dynamics of Rotors and Foundations, Springer-Verlag, Berlin, Heidelberg, New York, p. 383.

[2] S.S. Rao, 2005. Mechanical Vibration, SI Edition, Prentice Hall.

[3] G.K. Singh, S.A.S. Al Kazzaz, Induction machine drive condition monitoring and diagnostic research - /a survey, Electric Power Systems Research 64 (2003) 145-158.

[4] C.D. Foster, J.E. Mottershead, 1989. Parameter Estimation Techniques for Monitoring Machines and Structures, Modern Practice in Stress and Vibration Analysis, 109-124.

[5] V.T. Tran, B.-S. Yang, 2012. An intelligent condition-based maintenance platform for rotating machinery, Expert Systems with Applications, 39, 2977-2988

[6] P.F. Cavalcante & K.L. Cavalca, A Method To Analyse The Interaction Between Rotor-Foundation Systems, Proceedings of the 16 International Modal Analysis Conference (IMAC98), Santa Barbara, USA (1998) 775-781

[7] M. Yu, N. Feng, E. Hahn, 2012. Identification of foundations in rotating machinery using modal parameters, 10th International Conference on Vibrations in Rotating Machinery, 89-98

[8] D.B. Li, L.H. Yam, 2000. Modal synthesis method for vibration isolation design of massive rotating machines resili-

ently supported by an elastic structure, Journal of Sound and Vibration, 231, 233–245

[9] P. Bonello, M. J. Brennan, 2001. Modelling the Dynamic Behaviour of a Supercritical Rotor on a Flexible Foundation using the Mechanical Impedance Technique, Journal of Sound and Vibration, 239(3), 445-466

[10] K.L. Cavalca, P.F. Cavalcante, E.P. Okabe, 2005, An investigation on the influence of the supporting structure on the dynamics of the rotor system, Mechanical Systems and Signal Processing, 19, 157–174.

[11] R.O. Curadelli, R.D. Ambrosini, R.F. Danesi, 2004, Vibration control by attaching masses to a plate excited by rotating machinery, Journal of Sound and Vibration 273, 1087–1100.

[12] C. Devriendt, G. Steenackers, G. DeSitter, P. Guillaume, 2010, From operating deflection shapes towards mode shapes using transmissibility measurements, Mechanical Systems and Signal Processing, 24, 665-677.

[13] O. Dossing, and C.H. Staker, 1987, Operational deflection shapes: background, measurement and applications, 5th International Modal Analysis Conf., London, UK, 1372–1378.

[14] W.D. Marscher, and C.W. Jen, 1999, Use of operating deflection and mode shapes for machinery diagnostics, Proceedings of 17th International Modal Analysis Conference, 65–71

[15] P.L. McHargue, and M.H. Richardson, 1993, Operating detection shapes from time versus frequency domain measurements, The 11th International Modal Analysis Conference, Kissimmee, USA, 581-587

[16] R. Pascual, J.C. Golinval, and M. Razeto, 1999, On-line damage assessment using operating detection shapes, Proceedings of 17th International Modal Analysis Conference, 238–243.

[17] B.H. Tongue, Principles of Vibration, Oxford University Press, 2nd ed., 2002.

[18] S. N. Ganeriwala , V. Kanakasabai, A. Muthukumarasamy, T. Wolff, M. Richardson, 2011, Using Operating Deflection Shapes to Detect Faults in Rotating Equipment, Structural Dynamics Conference Proceedings, 3, 265-271

[19] S. N. Ganeriwala, Z. Li, M. H. Richardson, 2008, "Using Operating Deflection Shapes to Detect Shaft Misalignment in Rotating Equipment" Proceedings of International Modal Analysis Conference (IMAC XXVI): Conference & Exposition on Structural Dynamics.

[20] S. N. Ganeriwala, B. Schwarz, M. Richardson, 2012, "Using Operating Deflection Shapes to Detect Unbalance in Rotating Equipment" Proceedings of International Modal Analysis Conference (IMAC XXVII), February, 2009.

[21] K. Singleton, B. Bracher, 2012, Analysis of Fan Excessive Vibration Using Operating Deflection Shape Analysis - Case Study, Uptime Magazine, 58-59

[22] K.K. Kong S. Noroozi, A.G.A. Rahman, M. Dupac, H.C. Eng, Z.C. Ong, S.Y. Khoo, and J. Vinney, 2014, Non-destructive Testing and Assessment of Dynamic Incompatibility between 3rd Party Piping and Drain Valve Systems: An Industrial Case Study. Nondestructive Testing and Evaluation, 29, 154-163

[23] A. G. A. Rahman, S. Noroozi, M. Dupac, S. M. S. M. Al-Atta, and J. E. Vinney, 2013, A Hybrid Approach for Nondestructive Assessment and Design Optimisation and Testing of in-Service Machinery, Nondestructive Testing and Evaluation, 28, 44-57.

[24] D.L. Brown, R.J. Allemang, R. Zimmerman, M. Mergeay, 1985 Parameter estimation techniques for modal analysis, in: Seventh International Seminar on Modal Analysis, Katholieke Universiteit Leuven, Leuven, Belgium

[25] M.H. Richardson, D.L. Formenti, 1982, Parameter Estimation from Frequency Response Measurements Using Rational Fraction Polynomials: Proceedings of the 1st International Modal Analysis Conference, Orlando, Florida, 167–186.

[26] M.H. Richardson, D.L. Formenti, 1985, Global Curve Fitting of Frequency Response Measurements Using the Rational Fraction Polynomial Method, in: Proceedings of the 3rd International Modal Analysis Conference, Orlando, Florida, USA, 390–397

[27] B. Schwarz, M.H. Richardson, 2007, Using a De-Convolution Window for Operating Modal Analysis, in: Proceedings of the 2nd International Operational Modal Analysis Conference, Orlando, Florida, USA, 1-7.

[28] L.M. Zhang, R. Brincker, P. Andersen, 2001, Modal Indicators for Operational Modal Identification, in: Proceedings of the 19th International Modal Analysis Conference, Orlando, Florida, USA, 746-752

[29] A.G.A. Rahman, O. Z. Chao, Z. Ismail, 2011, Enhancement of coherence functions using time signals in Modal Analysis, Measurement, 44, 2112-2123

[30] A.G.A. Rahman, Z. Ismail, S. Noroozi, and Z.C. Ong, 2014, Enhancement of Impact-Synchronous Modal analysis (ISMA) with number of averages. Journal of Vibration and Control, 20,1645-1655

[31] Z.C. Ong, A.G.A. Rahman, Z. Ismail, and S. Norooz, 2013, Impact-Synchronous Modal Analysis (ISMA)- An Attempt to Find an Alternative, Proceedings of 5th IOMAC International Operational Modal Analysis Conference

[32] M. Dilena, A. Morassi, 2004, Experimental modal analysis of steel concrete composite beams with partially damaged connection, Journal of Vibration and Control, 10, 897-913

[33] M.M. Fayyadh, H.A. Razak, Z. Ismail, 2011, Combined modal parameters-based index for damage identification in a beamlike structure: theoretical development and verification, Arch. Civ. Mech. Eng., 11, 587-609.

[34] S. Noroozi, P. Sewell, A.G.A. Rahman, J. Vinney, O.Z. Chao, and B. Dyer, 2013, Modal analysis of composite prosthetic energy-storing-and-returning feet: an initial investigation. Proceedings of the Institution of Mechanical Engineers Part P: Journal of Sports Engineering and Technology, 227, 39-48

[35] M.J. Whelan, M.V. Gangone, K.D. Janoyan, R. Jha, 2011, Operational modal analysis of a multi-span skew bridge using real-time wireless sensor networks, Journal of Vibration and Control, 17, 1952-1963

[36] I. Zaghbani, V. Songmene, 2009, Estimation of machine-tool dynamic parameters during machining operation through operational modal analysis, International Journal of Machine Tools & Manufacture, 49, 947-957

[37] D.J. Ewins, Modal Testing: Theory, Practice and Application, Research Studies Press Ltd., England, Hertfordshire, Baldock, 2000

[38] H. Vold, B. Schwarz, and M. Richardson: 2000, 'Measuring operating deflection shapes under non-stationary conditions', Proceedings of the 18th International Modal Analysis Conference, Kissimmee, Florida

[39] B. J. Schwarz and M. H. Richardson: CSI Reliability Week, Orlando, FL, 1999

[40] D. Di Maio, N. Zampognaro, C. Zang and D. J. Ewins, 2006, ”Effect of vibration excitation locations on structural damage detection using the CSLDV technique: simulation and testing”, Proc. SPIE 6345, 63450O.

[41] J. Vinney, S. Noroozi, A.G.A. Rahman, P. Sewell, Z.C. Ong, K. K. Kong, and M. Dupac. 2012, Analysis of Composite Prosthetic Energy-Storing-and-Returning (ESR) feet: A comparison between FEA and the experimental analysis. International Journal of COMADEM, 15, 19-28.