Model Based Unbalance Identification for Paper Machine's Tube Roll

Choudhury Tuhin, Kurvinen Emil, Sopanen Jussi

This is a Final draft version of a publication
published by Springer, Cham
in Advances in Mechanism and Machine Science. IFToMM WC 2019.

DOI: 10.1007/978-3-030-20131-9_333

Copyright of the original publication: © Springer Nature Switzerland AG 2019

Please cite the publication as follows:

Choudhury T., Kurvinen E., Sopanen J. (2019) Model Based Unbalance Identification for Paper Machine's Tube Roll. In: Uhl T. (eds) Advances in Mechanism and Machine Science. IFToMM WC 2019. Mechanisms and Machine Science, vol 73. Springer, Cham. https://doi.org/10.1007/978-3-030-20131-9_333

**This is a parallel published version of an original publication.
This version can differ from the original published article.**

Model Based Unbalance Identification for Paper Machine's Tube Roll

Tuhin Choudhury*[0000–0003–0457–2391], Emil Kurvinen, and Jussi Sopanen

Department of Mechanical Engineering
Lappeenranta University of Technology
Skinnarilankatu 34, 53850 Lappeenranta, Finland
Tuhin.Choudhury@lut.fi Emil.Kurvinen@lut.fi Jussi.Sopanen@lut.fi

Abstract. Mass unbalance is a major concern in modern day rotating machinery. For single and double disc rotor bearing systems, model based techniques have been used to identify the unbalance. The aim of this research is to predict the unbalance in a similar way for a paper machine's tube roll using its virtual model. In this case study, the rotor is a large diameter continuous tube with thin walls and without any mounted disc. To identify the unbalance magnitude and phase, modal expansion and equivalent load minimization by least squares is used. The distributed equivalent load is sorted using two different methods. The first method considers only the isolated maximum load. In the second method, the load at all the nodes are taken into account. This first method is tested on a single disc rotor bearing system from literature and the unbalance parameters were predicted with good accuracy. However, for the paper machine's tube roll, due to its continuous structure, the equivalent load is distributed more evenly across the rotor. Therefore the second method is able to predict the unbalance parameters with better accuracy.

Keywords: Equivalent Load Minimization, Modal Expansion, Model-Based Identification, Paper Machine Roll, Unbalance.

1 Introduction

In the present day, rotating machineries are common in industrial applications. Such machines are prone to a number of faults which can be attributed to manufacturing limitations or fault in assembly. Mass unbalance is one such fault which is the most frequently occurring source of vibration [1]. Although rotating machines are balanced to specific balancing class, for e.g. according to ISO 1940-1 standard, there is always some amount of residual unbalance left in the rotating system. Such faults might be insignificant initially but may develop to vibrations of larger magnitude due to operational conditions such as heat generation, wear and looseness leading to premature breakdown of the system [2].

To prevent a critical impact due to unbalance, it is important to continuously monitor the system state. In modern rotating machinery, signal processing is combined with physics based modeling techniques to analyze vibration behaviour of the system. The occurrence of unbalance alters the system's vibration

measurements. This information is used in model based approach to identify the unbalance in its initial stage and determine its location and magnitude [2].

Over the last two decades, many researchers in the field of model based diagnostics used the concept of modal expansion and equivalent load minimization to determine the fault parameters. Markert et al. [3] used model based identification to predict different fault parameters including unbalance. They used modal expansion to obtain vibration based quantitative data for the entire system and combined it with the least squares method in time domain to identify the fault parameters. Bachschmid et al. [4] also used least squares method to identify multiple faults in a finite element (FE) based rotating system model by minimizing the vibration residuals in frequency domain. In a very similar way, Jain and Kundra [5] observed that the occurrence of fault alters the dynamic behavior of the system and utilized the equivalent load for identification of unbalance and transverse crack.

Sudhakar and Sekhar [2] proposed a method to improve the accuracy of measurement of unbalance based on modal expansion for a single disc rotor system. They used least squares method for the equivalent load minimization and as well as for the vibration minimization and identified the unbalance using measured transverse vibrations at one location only. Shrivastava and Mohanty [1] proposed a method of combining modal reduction, Kalman filter and recursive least squares method to determine the phase and amplitude of unbalance in a single plane. The process is considered as a suitable alternative to modal expansion once extended to multi-plane unbalance identification. More recently, Yao et al. [6] combined modal expansion along with inverse problem approach to eliminate the shortcomings related to low number of measurement points. They concluded that the combination provided more accurate results than purely modal expansion based estimations.

As evident from the existing literature, quite a few research papers in the field of model based identification used the concepts of modal expansion, equivalent load distribution and least square optimization to determine the position and magnitude for unbalance. However, to the best of the author's knowledge, most of these studies were conducted for single disc or double disc rotor-bearing systems. For such cases, the discs act as additional mass point leading to higher equivalent load concentration at the disc locations in the rotor system. Due to the high load, it is probable that the unbalance position is predicted to be at the disc location, irrespective of where the actual unbalance is located. However, for a continuous rotor with relatively more even distribution of mass, the prediction of the unbalance location can be tested adequately. The objective of this article is to apply model based identification for predicting mass unbalance in case of a paper machine's tube roll. For this case example, the rotor is a large diameter tube with thin walls without any mounted disc. Modal expansion and equivalent load distribution methods are used for the prediction.

2 Theoretical Background of Modal Expansion and Equivalent Load Minimization Based Method

In this method, the rotor-bearing-support system is created using finite element method. Timoshenko beam elements are used to design the rotor model. Since only lateral vibrations are considered for this study, the axial and torsional degrees of freedom (dofs) are constrained.

2.1 System Equations

For modelling purpose, the entire system of the rotating machinery is represented by its mass, damping and stiffness properties in the form of equation of motion. Using the FE model of the system, the system matrices are derived and all external forces are also considered, giving the equation of motion for the original undamaged system as:

$$\mathbf{M}\ddot{\mathbf{r}}_o(t) + (\mathbf{C} + \omega\mathbf{G})\dot{\mathbf{r}}_o(t) + \mathbf{K}\mathbf{r}_o(t) = \mathbf{F}_o(t) \quad (1)$$

where $\mathbf{M}$, $\mathbf{C}$, $\mathbf{G}$ and $\mathbf{K}$ are the mass, damping, gyroscopic and stiffness matrices of the original system and force $\mathbf{F}_o$ results in vibration, $\mathbf{r}_o(t)$. Although, the gyroscopic matrix is included in the equation of motion for the system, it has very limited effect in the low speed rotors studied in this research. The damping matrix $\mathbf{C}$ includes the damping coefficient of the bearings at their respective location. Internal damping of the rotor is not considered in this study. In coherence with real rotating machinery, it is assumed that the system has a low amount of residual unbalance after initial grade based balancing.

Over a period of operation, an additional unbalance occurs in the system which can be defined by a fault parameter β . The fault causes new vibrational behavior $\mathbf{r}(t)$ in the damaged system. These vibrations, when compared to the the original undamaged system, give a residual displacement, velocity and acceleration respectively as:

$$\delta\mathbf{r}(t) = \mathbf{r}(t) - \mathbf{r}_o(t) \quad (2)$$

$$\delta\dot{\mathbf{r}}(t) = \dot{\mathbf{r}}(t) - \dot{\mathbf{r}}_o(t) \quad (3)$$

$$\delta\ddot{\mathbf{r}}(t) = \ddot{\mathbf{r}}(t) - \ddot{\mathbf{r}}_o(t) \quad (4)$$

These residuals are imagined to be caused by additional theoretical equivalent force $\Delta\mathbf{F}(\beta, t)$ acting on undamaged system instead of fault β . Therefore, correlating the residual parameters with the equivalent forces results in the following equation of motion:

$$\mathbf{M}\delta\ddot{\mathbf{r}}(t) + (\mathbf{C} + \omega\mathbf{G})\delta\dot{\mathbf{r}}(t) + \mathbf{K}\delta\mathbf{r}(t) = \Delta\mathbf{F}(\beta, t) \quad (5)$$

2.2 Modal Expansion

Equation (5) shows that the equivalent loads $\Delta\mathbf{F}(t)$ generated in the system are estimated using measured vibration response at all degrees of freedom (dof) of the system. However, it is not economical and also not feasible to measure

vibration response at all the dof. Thus the vibration response is measured only at a few designated dof. For m number of measured dofs, the residual displacements at all coordinates, $\delta \mathbf{r}(t)$ can be correlated to those at measured location $\delta \mathbf{r}_m(t)$ by a measurement / mapping matrix $\mathbf{A}$.

$$\delta \mathbf{r}_m(t) = \mathbf{A} \delta \mathbf{r}(t) \quad (6)$$

The mapping matrix $\mathbf{A}$ is of size $m \times n$ and measured locations have a value of one and rest are zeros. In order to approximate vibrations at unmeasured dofs, the mode shape matrix, Φ is used along with the measured vibrations. This process is referred to as modal expansion and by applying least squares method for matrices, the residual vibrations at all locations can be obtained as:

$$\delta \mathbf{r}(t) = \Phi \left[(\mathbf{A} \Phi)^T (\mathbf{A} \Phi) \right]^{-1} (\mathbf{A} \Phi)^T \delta \mathbf{r}_m(t) \quad (7)$$

2.3 Equivalent Load and its Reordering Procedure

Based on equation (7), the vibration residuals at all coordinates are estimated. These residuals are simply combined with the original system matrices in the form of equation of motion to obtain the equivalent load.

$$\Delta \mathbf{F}(t) = \mathbf{M} \delta \ddot{\mathbf{r}}(t) + (\mathbf{C} + \omega \mathbf{G}) \delta \dot{\mathbf{r}}(t) + \mathbf{K} \delta \mathbf{r}(t) \quad (8)$$

The equivalent load obtained from equation (8) is a direct representation of the effect of the unbalance on the rotor system. In the previous cases, the equivalent load values are found to be distributed at all the locations of a rotor with highest magnitude at fault location [2, 3]. However, for a single unbalance, the theoretical forces act only at the location of the unbalance with no component at other locations. Therefore, for closer representation to the theoretical unbalance forces, two different methods are used. The first method considers the maximum load in vertical and horizontal directions at the location of unbalance nodes and while the values at all other nodes are ignored [6]. In the the second method, the equivalent loads are shifted from all nodes to the particular node with highest magnitude and the summation of all the nodes is used for minimization.

2.4 Fault Model and Fault Parameters in Case of Mass Unbalance

Next, as the unbalance location is predicted, the fault parameters to be determined are the eccentricity and phase. For this purpose, fault models are created which are theoretical representation of forces generated due to unbalance. For a single unbalance fault β of magnitude u_n and phase ϕ_n occurring at location n of a rotor, the equivalent forces acting in vertical and horizontal directions can be denoted as:

$$\Delta \mathbf{F}_v(\beta, t) = \omega^2 u_n [\sin(\omega t + \phi_n)] \quad (9)$$

$$\Delta \mathbf{F}_h(\beta, t) = \omega^2 u_n [\cos(\omega t + \phi_n)] \quad (10)$$

The unbalance parameters are determined by minimizing the difference between the theoretical forces $\Delta \mathbf{F}(t)$ and the equivalent loads due to unbalance $\Delta \mathbf{F}(\beta, t)$. Often the least squares optimization method is used for this purpose [2, 7, 6].

$$\int \left| \Delta \mathbf{F}(\beta, t) - \Delta \mathbf{F}(t) \right|^2 dt = \text{Min} \quad (11)$$

The optimization algorithm will adapt the fault parameters from a given initial value until the deviation between the theoretical unbalance forces and the equivalent loads reaches a predefined minimum. Overall, the method has a limitation that the error in identified fault parameters increases with decrease in number of measured vibrations [6].

3 Implementation in Test Case

The aim of the paper is to develop an unbalance detection algorithm for a continuous rotor, where there are no mounted discs. For this purpose a paper machine's steel tube roll is used as the test rig.

3.1 Test Rig Description and Simulation Model

Fig. 1 shows a schematic of the paper machine's tube roll while Table 1 describe its physical properties respectively in SI units.

Fig. 1. Schematics of Paper machine's tube roll. All dimensions are in meters.

Table 1. Parameters of Paper Machine's tube roll rotor bearing system

Rotor material properties	
Density	7800 kg/m ³
Poisson's ratio	0.3
Young's modulus	$2 \cdot 10^{11}$ Pa
Bearing properties	
Bearing stiffness coefficient	
Vertical	$3 \cdot 10^8$ N/m
Horizontal	$2 \cdot 10^8$ N/m
Bearing damping coefficient	
Vertical	$3 \cdot 10^3$ Ns/m
Horizontal	$4 \cdot 10^3$ Ns/m

The FE model of the rotor-bearing-support system consists of 24 beam elements separated by 25 nodes along its length (Fig. 1). Node 2 and 24 are the bearing locations. The support locations are exactly the same as respective bearing locations although they are represented by nodes 26 and 27. The support

structures contribute to an additional mass of 127 kg each at respective location. Other than these, the overhanging part from the tube end is also considered as an additional mass point of approximately 6 kg at node 6 and node 20 respectively. Lastly, after initial balancing of the tube roll, the balancing planes are identified at node 6 and 20 with a balancing mass of 4 kg each.

4 Results and Discussions

The results from two cases are demonstrated in this paper. First, the verification of the functionality of the algorithm is carried out using a test case from literature and then the algorithm is applied for the tube roll.

4.1 Algorithm Verification by Using a Literature Based Test Case

To verify the accuracy of the algorithm, a test case from literature is considered which consists of a single disc-rotor-bearing-system [6]. For simulation purpose, the properties of the rotor and bearing parameters as described by Yao et al. are taken into account. The system matrices are acquired using the recreated FE model of the rotor (Fig. 2).

Fig. 2. Recreated rotor model from Yao et al. [6] test case.

The critical speeds of the recreated model are compared to the actual Yao et al. model in Table 2 which shows that the second set of mode shapes are not visible for the recreated model. This might be due to asymmetry in the original model which could not be recreated with the available information. However, the first and the third frequencies for the recreated model are quite close to the original model used by Yao et al. Therefore, calculation are carried out for the rotor-bearing system with numerically simulated measured vibrations. The

Table 2. Comparison of first three critical speeds in Yao. et al [6] case

Critical speeds (Hz) at 6000rpm		
Mode	Yao et al.	Recreated model
1	46.5	45.5
2	46.7	45.5
3	50.0	-
4	50.3	-
5	313.9	300.8
6	358.8	343.4

recreated model was tested with four different sets of unbalance with different magnitude and phase respectively. The equivalent load distribution (Fig. 3(a)) for the recreated model with four measured dofs (maximum load 15.12 N at node 7) closely resembles to the actual load distribution obtained by Yao et al. (maximum load 15.22 N at node 7). The equivalent load minimization is performed by using least squares optimization method unlike the original article. The results from the recreated model are compared to the results obtained by Yao et al. using optimization algorithm Antlion in Fig. 3(b). The actual unbalance is also plotted in the figure for reference and the result from different cases are displayed in Table 3. The result shows that the current algorithm is fairly accurate in its unbalance prediction as compared to the cases in literature.

Fig. 3. (a) Equivalent vertical load of an unbalance of 45 g·mm at 135 ° obtained by recreated model shows max load at node 7 (15.12 N). (b) Predictive Model Comparison: Recreated model vs Yao et al. antlion algorithm.

Table 3. Comparison of actual and predicted unbalance parameters: recreated model vs Yao et al. [6] antlion algorithm (ALO).

Test case	Unbalance magnitude (g·mm)					Unbalance phase (degree)				
	Actual	Recreated model		Yao et al's ALO		Actual	Recreated model		Yao et al's ALO	
		Predicted	Error %	Predicted	Error %		Predicted	Error %	Predicted	Error %
1	29.40	25.02	14.89	25.05	-14.80	45	45.29	0.65	45.27	0.60
2	39.30	33.46	14.85	33.48	-14.81	180	180.00	0	180.31	-0.36
3	45.00	38.30	14.89	38.34	-14.80	135	134.71	-0.21	135.27	0.20
4	49.20	41.87	14.89	41.92	-14.80	225	225.29	0.13	225.25	0.11

4.2 Paper Machine's Tube Roll

For the initial testing, the measurement data are simulated numerically based on the number of measurement locations. The measured signal is assumed to be obtained from vertical and horizontal directions only, as it would be obtained with, for example, displacement sensors such as hall sensors. The study is considered for a single period of rotation at the operational speed of 1200 rpm. After balancing the tube roll using ISO 1940-1:2003 recommended balancing quality

grade of G 1.0, the permissible residual unbalance for the non-damaged system is 0.0037 kg·m. This unbalance is assumed to be located at node 7 since the mass concentration is higher at that particular location for the tube roll.

Over a period of operation at constant speed, the residual unbalance at node 7 is assumed to increase up to 0.2 kg·m at 45 degree phase angle. For this single case, the distributed equivalent load across the rotor nodes is shown in Fig. 4(a). Similar to the recreated test case from literature, the maximum equivalent load is isolated and forces at all other nodes are ignored. Fig. 4(b) shows the the isolated maximum equivalent load.

Fig. 4. (a) Equivalent load distribution throughout the rotor (b) Isolated maximum equivalent load

Prediction Based on Maximum Equivalent Load Only. Before proceeding for the optimization of the isolated maximum equivalent load from Fig. 4(b), the theoretical equivalent load is calculated using equations (9) and (10) and a reference plot, showing how the actual unbalance forces would look like, is created for comparison (Fig. 5(a)).

Fig. 5. (a) Actual vertical load due to unbalance of magnitude 0.196 kg·m and phase 0 (b) Equivalent load from all nodes shifted to location of maximum value.

By comparing the values from Fig. 5(a) and 4(b), it can be established that the isolated highest equivalent load is approximately 6 times lower than the

actual load. Therefore, even with least square optimization, the predictions obtained are quite inaccurate for the tube roll even though good predictions were obtainable for the literature test case. The reason behind this could be that the static weights are higher in the paper machine's roll and the distribution of equivalent load is spread across the tube roll due to more uniform distribution of mass unlike the literature test case where the mass is concentrated at the disc location. Therefore, a different method should be used in this case which takes into account the equivalent load components at the other nodes as well.

Prediction Based on Shifting the Equivalent Load from All Nodes. In this method, the vertical and horizontal components of equivalent loads at all nodes are shifted to their corresponding location of maximum value. The values in the location of rotational dofs are ignored since the unbalance components considered are only vertical and horizontal in direction. Additionally, values in the support nodes are ignored since equivalent loads are derived from unbalance which does not affect the supports directly. Fig. 5(b) shows the shifted equivalent load from the same distribution in Fig. 4(a).

This shifted equivalent load closely represents the magnitude of the actual unbalance force in vertical direction shown in Fig. 5(a). However, the force progression is out of phase from the reference case. A simple phase correction step is included to reset the phase of the equivalent load to zero, i.e. same as the theoretical actual load. Next, the finalized equivalent load is optimized using least squares method for prediction of unbalance parameters. Prediction results are simulated for different number of measured coordinates to analyze how the accuracy varies with measured data. Table 4 shows the results obtained for 50, 20, 16, 8, 4 and 2 measured dofs respectively. The results in Table 4 shows

Table 4. Unbalance magnitude and phase estimation for steel tube roll using equivalent load minimization with different number of measured coordinates.

Test no.	No. of measured dofs	Node	Unbalance magnitude (kg.m)			Unbalance Phase (degree)		
			Actual	Estimated	Error %	Actual	Estimated	Error %
1	50	7	0.19	0.19	0.61	45	45.81	1.81
2	20	7	0.19	0.19	0.73	45	45.82	1.82
3	16	7	0.19	0.19	1.25	45	45.88	1.96
4	8	7	0.19	0.15	20.02	45	48.50	7.78
5	4	7	0.19	0.15	19.37	45	50.46	12.14
6	2	7	0.19	0.13	30.89	45	57.58	27.97

the definitive trait of modal expansion where the prediction deteriorates with decrease in number of measured coordinates [3]. However, the prediction for unbalance magnitude seem to improved overall after considering shifting of load values from other nodes instead of calculating with just the maximum values. A comparison of prediction accuracy between the two methods along with the actual unbalance for reference is shown in Fig. 6.

Fig. 6. Comparison of predicted unbalance magnitude with actual magnitude.

5 Conclusions

Overall, the study shows that modal expansion and equivalent load minimization methods can be used for predicting mass unbalance in case of large industrial rotor such as the paper machine's roll. However, the method of isolating the equivalent load did not yield desirable results for the paper machine's tube roll simulated as a rotor-bearing-support system (84.96 % error in predicted unbalance magnitude with measurements from bearing locations). On the other hand, by taking the forces at all the locations into account, better predictions were achieved (19.87 % error). The unbalance location was correctly predicted for cases when the unknown unbalance was simulated at the end points of the tube shell (Node 7 and 19). However, further research needs to be conducted on different realistic rotor prototypes to understand how to improve the accuracy of unbalance prediction, especially for low number of measured coordinates (2 or 4 dofs).

References

- [1] Shrivastava A, Mohanty A (2018) Estimation of single plane unbalance parameters of a rotor-bearing system using kalman filtering based force estimation technique. *Journal of Sound and Vibration* 418:184–199 doi: 10.1016/j.jsv.2017.11.020
- [2] Sudhakar G, Sekhar A (2011) Identification of unbalance in a rotor bearing system. *Journal of Sound and Vibration* 330(10):2299–2313. doi: 10.1016/j.jsv.2010.11.028
- [3] Markert R, Platz R, Seidler M (2001) Model based fault identification in rotor systems by least squares fitting. *International Journal of Rotating Machinery* 7(5):311–321. doi: 10.1155/S1023621X01000264
- [4] Bachschmid N, Pennacchi P, Vania A (2002) Identification of multiple faults in rotor systems. *Journal of sound and vibration* 254(2):327–366. doi: 10.1006/jsvi.2001.4116
- [5] Jain J, Kundra T (2004) Model based online diagnosis of unbalance and transverse fatigue crack in rotor systems. *Mechanics Research Communications* 31(5):557–568. doi: 10.1016/j.mechrescom.2003.11.002
- [6] Yao J, Liu L, Yang F, Scarpa F, Gao J (2018) Identification and optimization of unbalance parameters in rotor-bearing systems. *Journal of Sound and Vibration* 431:54–69. doi: 10.1016/j.jsv.2018.05.050
- [7] Sekhar A (2005) Identification of unbalance and crack acting simultaneously in a rotor system: Modal expansion versus reduced basis dynamic expansion. *Modal Analysis* 11(9):1125–1145. doi: 10.1177/1077546305042531