Dynamic Analysis of Gantry Crane Pillar Subjected to Industrial Base Excitation

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Abstract: Recently, the high-tech industry has become a key industry for economic development in many countries. Reliability, safety, efficiency and performance of machinery in all industrial applications are the main concerns. Gantry cranes are widely used in many industrial and civil applications. However, these cranes are vibration sensitive and are vulnerable to base excitation produced by the industrial equipment in vicinity, which may lead to heavy economical loss. In recent years, many experimental studies have been performed using shaking table tests to determine seismic response of structural models subjected to various earthquake records. In this study, a prototype is prepared of a particular gantry crane and shaking table with comprehensive calculations and the base excitation is artificially produced by the means of vibromotors. The paper elucidates the analysis of dynamic motion behaviour of the prototype using FFT Response, Time Domain Response and Orbit Plot tools, to understand the effect of industrial base excitation on the gantry crane and to provide a better layout for the setup of various machinery in the industry.

Keywords- shaking table, base excitation, dynamic motion behavior, FFT or Orbit Plot, modal analysis;

1. INTRODUCTION

Nowadays high-performance large size material handling and conveying machines, such as container cranes, huge gantry cranes, ship unloaders and ship loaders, etc., as well as construction and surface mining machines, such as e.g. bucket wheel excavators, spreaders, reclaimers, stacker feeding bridges etc., have found an extremely wide application in almost all areas of humanrunning life activities[1].Overhead cranes are widely used to move the large/heavy objects horizontally for either manufacturing or maintenance applications in many industrial environments, such as ocean engineering, nuclear industries, and airports, etc. [6]. Also, there is a continuous scope for improvement in efficiency and performance of such systems. The effect of seismic excitation on these systems have previously been studied, C. OktayAzeloglu [2] studied seismic behaviour of container cranes while Ahmet Sagirli [3] investigated dynamic behaviour of cranes under impact of seismic excitation. Also the use of shaking table test has been remarkable for understanding structural as well as dynamic behaviour of cranes and buildings [4-5].However, the effect of industrial base excitation is highly dubitable and is always neglected and the focus is drawn upon improving the efficiency and reducing the effect of earthquake tremors.

Here, a prototype model of gantry crane has been prepared by using a 1:10 scale ratio. Scale ratio of 1:20 was used by C.Oktay Azeloglu, Ahmet Sagirli [7] and the scaling method used here is according to Hamid Reza Tabatabaiefar and BitaMansoury [8]. Further, shaking table tests are performed with the use of digital analyser to understand the dynamic motion behaviour under the impact of artificial base excitation. Hong-Soek Park and Ngoc-Tran Le [9] researched on virtual prototype by simulating the real behaviour of harbour crane. Modal analysis on giant shipbuilding crane is also performed by Guojian Huang, Chengzhong He and Xinhua Wang [10] and M.L. Chandravanshi and A.K. Mukhopadhyay compared results of theoretical analysis and modal analysis.

In this review, modal analysis of a scaled prototype is performed to figure out the natural frequency of the setup to confirm the validity of the scaling method and also to assist in drawing conclusion. All these methods together derive a better pattern for arranging the rotating machinery in industries using gantry cranes to minimize the effect of base excitation on the crane. Although noticeable work has been done in the field of shaking table and studying gantry crane, very little work is reported which combines the use of scaling and prototyping in addition to modal analysis for understanding the proper dynamic motion of Gantry crane.

2. COMPUTATIONAL MODELLING AND MODAL ANALYSIS

The proposed system is schematized in Fig 1. A prototype of E-series Gantry Crane is designed by scaling an original model manufactured by SPANCO Inc. [11]

Knowing the required characteristics of the model, its three-dimensional numerical model has been created in SOLIDWORKS software using two-dimensional shell elements to model columns and beams as shown in Figure 1. The numerical model consists of a hoist beam, 2 square columns, 4 square pipe gussets, 4 castors with mounts and 2 steel plates. The material used for fabrication of crane is mild steel having density $7.70\frac{g}{cm^3}$.The dimensions of the crane have been determined in design process after taking a scaling ratio of 1:10 in order to fit the required natural frequency and mass. After the numerical modelling and design, detail drawings were prepared to reflect the design requirements of the prototype.

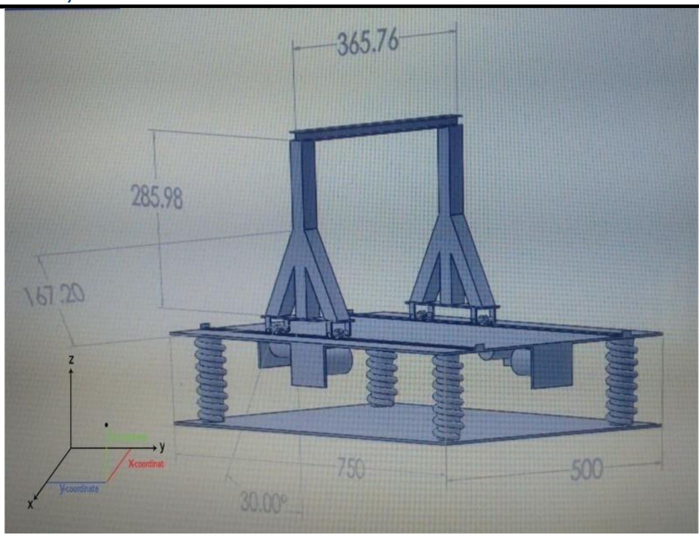

Fig 1 Computational model

Moreover, modal analysis of the setup was performed in ANSYS software to figure out the natural frequency. As illustrated in Fig 2 the natural frequency of the setup in mode 6 which is for the deformation in up and down swivelling motion of the table is 33.33 Hz.

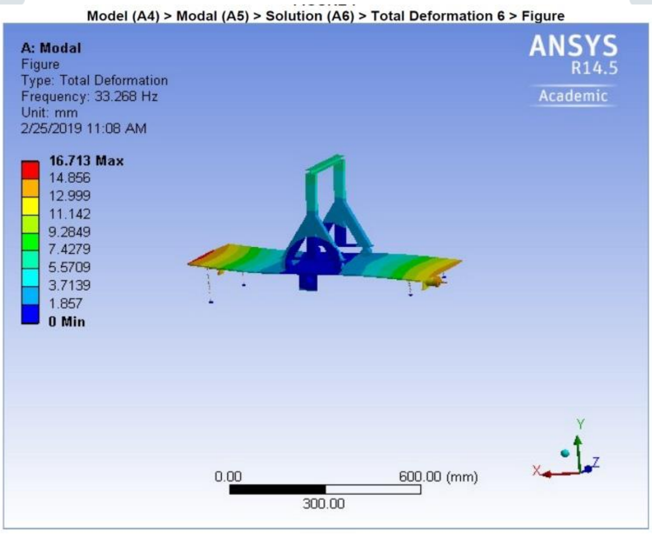

Fig 2 Modal Analysis

3. DEVELOPMENT OF EXPERIMENTAL SETUP

Construction details and arrangement of the prototype are illustrated in Fig 3. Thereafter, the detailed drawings were passed on to the engineering workshop where the steel plates and columns were cut and drilled according to the construction detailed drawings and then were welded together and the castors were mounted at equal distance on the bottom steel plates.

In the shaking table phase, a mild steel plate of 5mm thickness was procured and set as top plate while a wooden plate of 19mm thickness was considered as the base plate. Afterwards, the plates were assembled using 4 stainless steel springs (stiffness K=5.38 KN/m) at each corner using metal screws with 5-mm diameter and washers. Henceforth, to produce vibrations on the table, 4 D.C. motors with unbalanced weight on their shaft were clamped at the ends of 2 centerlines passing through the middle of the table. The maximum speed of the motors was 1000 r.p.m. and a D.C. controller was connected with them to regulate their speed. The mass of the model, without the wooden base plate was measured to be 20kg.

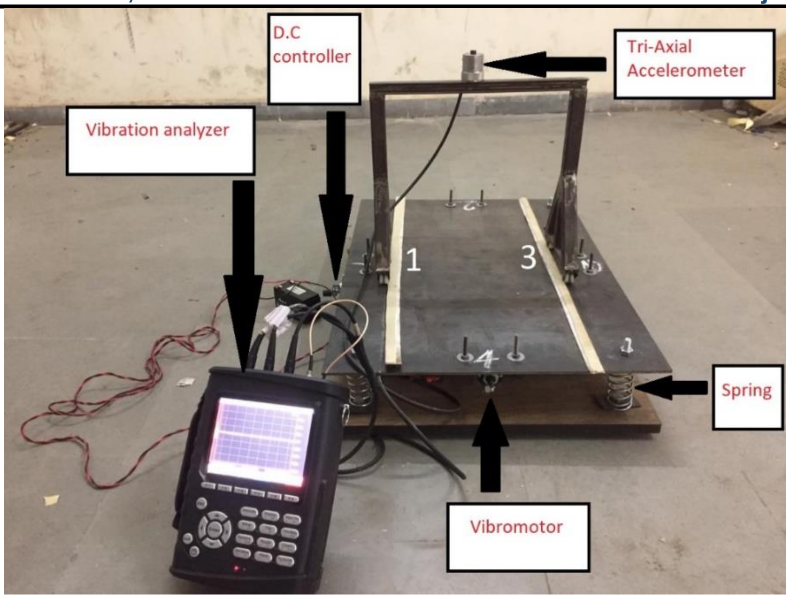

Fig 3Experimental setup

Further, the industrial crane experiences base excitation from every direction and hence to create a similar situation the motors situated underneath the top plate of shaking table are switched on and off in various combination. As shown in Fig 1 the direction along the length of the table is X direction and the direction along the width is Y direction. Motors are operated in sequence (1,3), (2,4) and (1,2,3,4) and the speed is varied with the controller. Data acquisition has been done using latest Vibration analyzer Coco80 which consists of one piezoelectric accelerometer (tri-axial) used for picking up the vibration signals from various stations on the test rig. This special piezoelectric pickup type of sensor has frequency range of 1-30 KHz, measurement range +/- 500g peak, resolution of 0.005 g and resonant frequency of 70 KHz. Both the sensor and vibration analyzer were installed on the set-up fabricated and a trial run was done. Vibration responses were acquired and analyzed by the analyzer with 3 input channels and sampling rate of 3.2 KHz.

The accelerometer is firstly placed on the crane top and then on the table top and the vibration response analysis of developed shaking table with prototyped crane at different operating conditions have been analyzed using FFT, Time Response and Orbit Plot.

4. RESULTS AND DISCUSSION

Data was gathered from the digital analyzer and FFT, Time Response and Orbit Plots for the crane and the table were prepared for 3 different combinations of the vibromotors. The 3 different combinations for generation of base excitation are,

1) Motors 1 and 3 are on,
2) Motors 2 and 4 are on,
3) Motors 1,2,3,4 are on.

Amplitude in x and z direction along with significant FFT response for each modulated base excitation frequency produced when probe is placed on the crane for combination 1 (1,3) is depicted in Table 1.The modulated frequency (Mf) is the summation of the natural frequency of prototype with the rotating frequency. Calculation of rotating frequency is based on the equation,

$f = \frac{N}{60}\ Hz$ where N=r.p.m.

Table-1 Case study when probe is on the crane for position (1,3)

MODULATED BASE EXCITATION FREQUENCY (Mf) (Hz)	AMPLITUDE (DISPLACEMENT) (mm)		SIGNIFICANT FFT RESPONSE (Hz)	
	x-direction	*z-direction*	*x-direction*	*z-direction*
50	18.3, -18.7	37.9, -37.3	21, 39, 92	21, 39, 92
46.66	12.4, -13.8	38.5, -41.9	19, 53, 98	19, 56, 201
41.66	8.91, -11.4	17.5, -16.7	11, 22, 91	11, 22, 44
36.66	13.8, -14.4	21.3, -19.8	20, 43, 90	43, 88, 170

The FFT, Time Response and Orbit Plot graphs for each modulated base excited frequency were developed and after thorough analysis several results were prepared. FFT and Time Response in X direction at Mf of 50Hz are illustrated in Fig 4 and 5 respectively. It is evident from Fig 4 that the sidebands are clearly visible. Also, the trend followed in Z direction for the same situation is similar to this trend which can be confirmed from the Table 1 as well. Fig 6 is the FFT at Mf of 36.66Hz in X direction. The maximum peak occurs at much high frequency which is a clear indication of abrupt chaotic motion of the system under this Mf. Moreover, the orbit plot in the Fig 7 displays a scattered representation of the points which is unevenly distributed in 4 quadrants and hence it supports the previous result that the system is undergoing a chaotic motion.

Observations of the system operating in combination 2 and 3 showed a similar response of the system under the impact of lower modulated base excitation. However, the same cannot be said for 50Hz Mf because the responses were not equivalent and differed as shown in Fig 8.

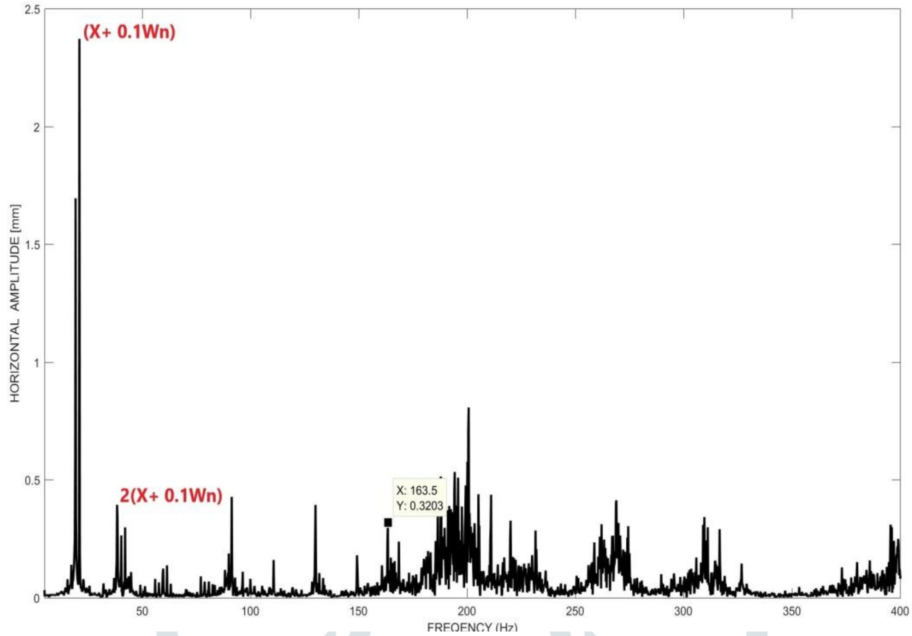

Fig 4 FFT in x direction @ 50Hz Mf

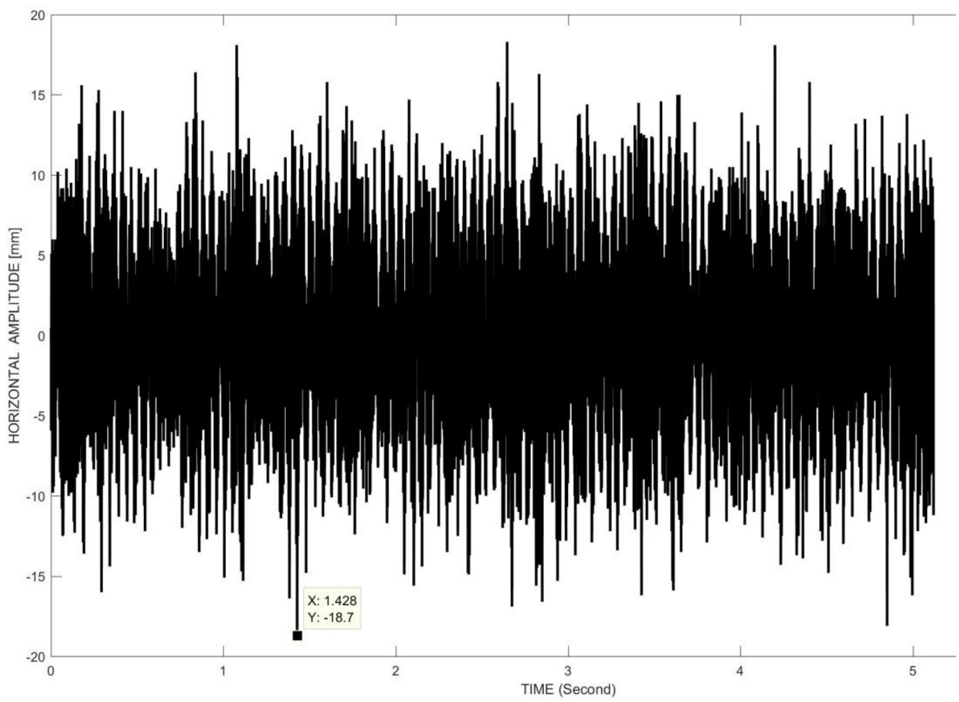

Fig 5 Time Response

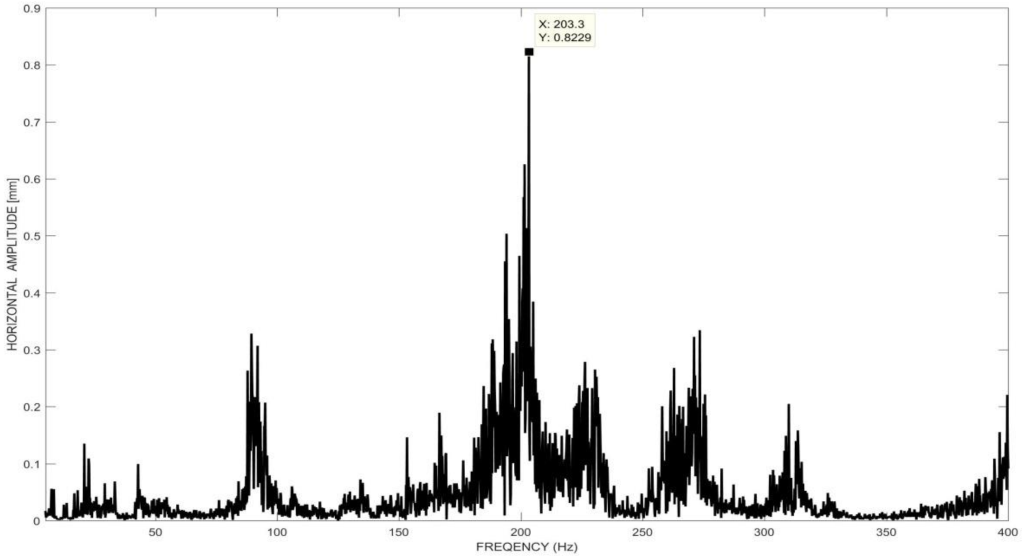

Fig 6 FFT in x direction @36.66Hz Mf

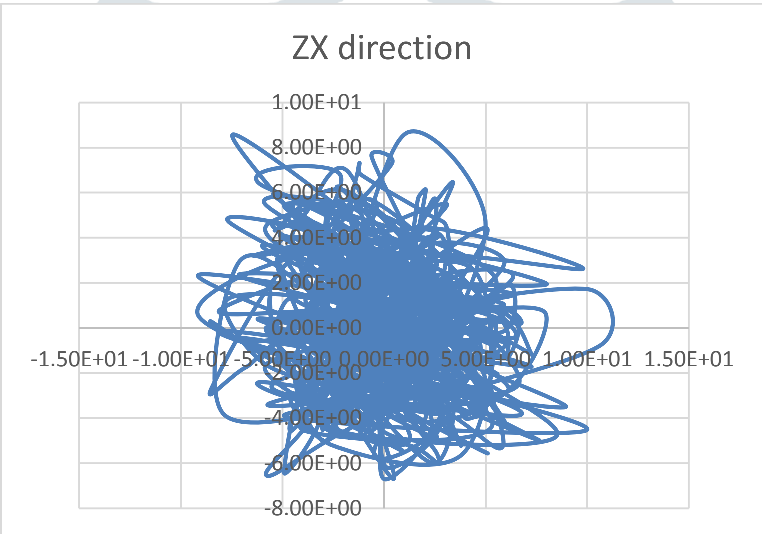

Fig 7 Orbit Plot @36.66Hz Mf

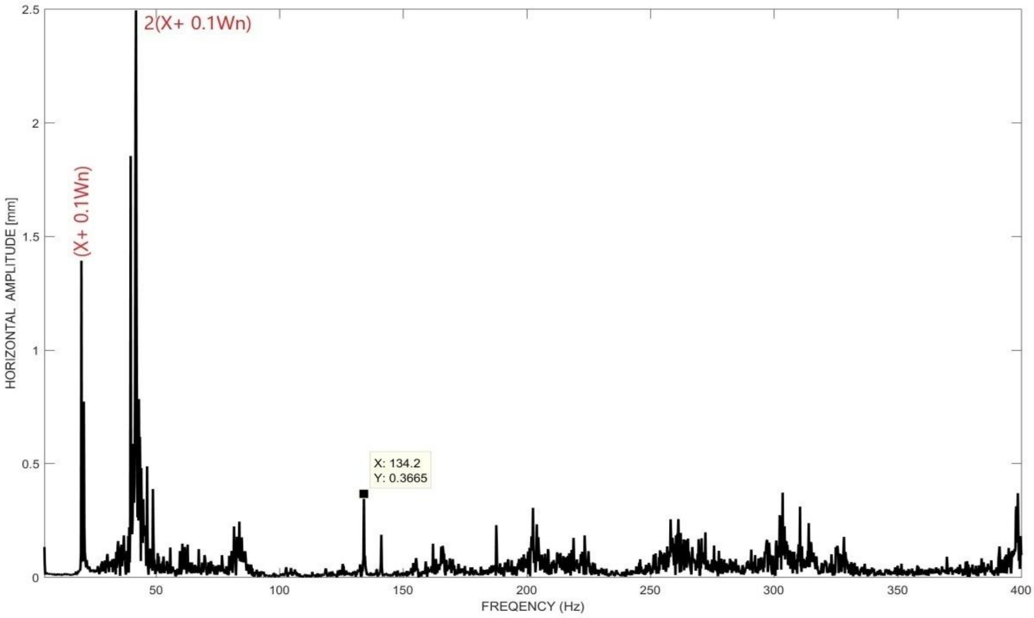

Fig 8 FFT @50Hz Mf for combination 2 (2,4)

5. CONCLUSION

In present research, effect of base excitation by machine tool has studied experimentally. A prototype of gantry crane was prepared along with shaking table. Controlled base excitations were induced in system by means of four vibromotor. Base excitations were produced by changing operating order number of vibromotors and corresponding data was acquired from base plate and top of crane for dynamic motion analysis. Significant outcomes of the present work are as follows:

1. Significant response is observed at (X+ 0.1Wn) Hz and it is harmonic.

2. From above relation (X+ 0.1Wn), value of the base excitation can be determined. Where, Wn is the natural frequency of the structure and X is the base excitation frequency.

3. It is recommended to design or maintain the structure to adjust the base excitation frequency above 8.33Hz to 16.67Hz.

4. Base excitation frequency of 13.33 Hz will produce more chaotic response of system and hence it should be avoided for overall stability of gantry crane at work shop.

5. Experimental study clearly led to conclude that all the heavy machineries must be arranged in such a way that line of axis of resultant base excitations should be parallel to the track of gantry crane. As observing the FFT response in combination 1(1,3) @modulated frequency of 50Hz has shown super harmonic motion response, while the FFT response in combinations 2(2,4) and 3(1,2,3,4) @modulated frequency of 50Hz has shown super harmonic route to chaotic behaviour.

6. This research work will be useful to project engineers to design the work shop layout for stable and efficient gantry crane operations.

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