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Design Dynamic Vibration Absorber and Experimental Testing for Reducing Vibration on Condensate Extraction Pump at Paiton Power Plant

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***Abstract* - One of the problems with the pump at Paiton Power Plant is the vibration. Vibration occurs because the pump's natural frequency approaches its operational frequency, then resulting condition in resonance. If the pump is experiencing resonance, there will be a strengthening of the vibration which can cause damage to the components in it. The vibration absorber was chosen as a way to reduce the vibration. The purpose of the absorber design is to determine the constants and mass of the absorber so that the mass displacement of the equipment experiencing vibration becomes zero. Vibration absorber installation experiment was carried out to prove that there was a change in the natural frequency for the system installed with the absorber. Installation of the Vibration Absorber can reduce the vibration of the pump due to resonance up to 75% at the desired frequency.**

Keywords: resonance, vibration, vibration absorber, natural frequency shifting.

I. INTRODUCTION

Vibration is a problem that often occurs in rotating equipment. This problem can be caused by several factors, namely misalignment of the shaft, vibration due to resonance, cracks in the rotor or shaft, rotor mass unbalance, loose bolt connections.

The current problem that occurs in the Paiton Power Plant's equipment which will be discussed in this paper is the vibration that occurs in the Condensate Extraction Pump (CEP). The pump is a type of centrifugal pump with an upright shaft position and function to pump fluid from the condenser into deaerator.

Problem vibration on CEP has been happening for last 3 months. The last data collection of vibration reached 7 mm/s (Fig.1). The vibration level by referred to the ISO 10816-3 Standard included in the ALARM category, which means that the pump cannot be run continuously and there is a possibility that it will cause catastrophic damage if it continues to operate under this condition [1].

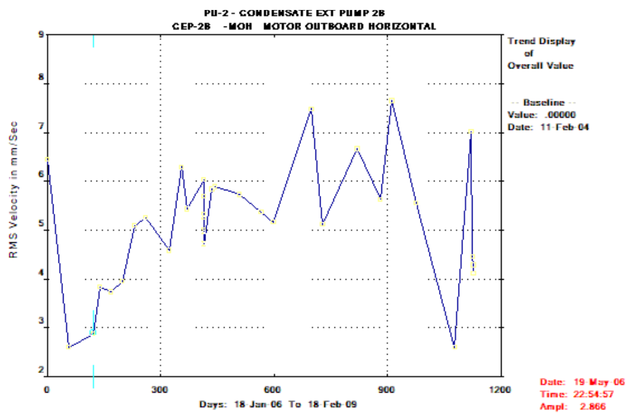

Figure 1: Vibration Trends on CEP

Vibration trends in Fig. 1 above are dominant at the 1x pump order frequency. It can be temporarily concluded that the excitation source is the residual unbalance of the rotor while the higher vibration in the horizontal direction is caused by the natural frequency in that direction being close to the operating frequency of the pump, resulting in resonance which causes amplification of the vibration in that direction.

There are several factors that affect the natural frequency of vertical pumps: weight of equipment height of construction above the deck, length of construction below the deck foot print [2]. To avoid this, it is necessary to change the natural frequency of operating frequency. Some options that can be done to solve the resonance problems follows Added stiffener to increase systems rigidity, pump's foundation repair, foot print repair, base plate replacement, changing operating frequency, inserting soft material, shifts natural frequency by increasing mass, and add vibration absorber on pump [2].

Considering the problems mentioned above, this study was conducted to reduce the vibration that occurs with a more efficient method by designing and applying a vibration absorber to the equipment.

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II. VIBRATION ABSORBER DESIGN

The first step in the design is the selection of a design to get the natural frequency, the determination of the material and the dimensions of the stick. After obtaining the dimensions and materials, then a simulation of the strength of the material is carried out to ensure that the selection of materials and dimensions is correct: Next, the absorber working frequency range is calculated, if the frequency is appropriate, the design calculation is correct, but if the frequency is not appropriate, the design calculation begins again from the first step.

The vibration absorber to be used is a cantilever beam with a length of **L** and mass of $\mathbf{w_1}$ then **a** is a cross-sectional area of **b**x**h** with a concentrated mass of $\mathbf{w_2}$ at a distance a as shown in Fig. 2 below. The design is not only simple and easy to make but also aims to get a natural frequency of 1500 rpm according to the frequency of the excitation force.

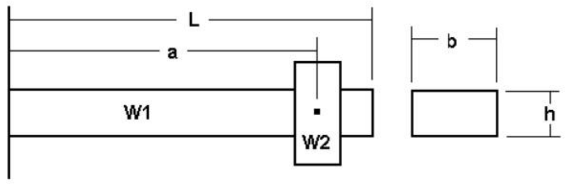

Figure 2: Vibration Trends on CEP

The natural frequency of the above system is formulated as follows:

$$\omega = \sqrt{\frac{g}{\delta w_1 + \delta w_2}} \qquad \text{(Eq.1)}$$

Where:

$$\delta w_1 = \frac{w_1 L^4}{8EI}$$ = static deflection as a result of beam weight w_1

= static $$\delta w_2 = \frac{w_2 a^2}{6EI}[3L - a]$$ deflection as a result of beam mass w_2

By entering δw_1 dan δw_2, into the equation 1 then,

$$\omega_{(rad/sec)} = \sqrt{\frac{6.EI.g}{w^2(3a^2L - a^3) + 0,75.w_1.L^4}} \qquad \text{(Eq.2)}$$

or, $$Nf_{(rpm)} = \sqrt{\frac{(2.114x10^5)E.I}{w_2(3a^2L - a^3) + 0.75 w_1 L^4}} \qquad \text{(Eq.3)}$$

So, the mass w_2 can be found with the equation

$$w_2 = \left(\frac{(2.114x10^5)E.I}{Nf^2.(3a^2L - a^3)}\right) - \frac{0.75 w_1 L^4}{3a^2L - a^3} \qquad \text{(Eq.4)}$$

Where:

- Nf : pump frequency (rpm)
- w_2 : concentrated mass weight (lbs)
- w_1 : beam weight per unit length (in)
- L : beam length (in)
- a : the position of the lumped mass on the beam (in)
- I : inertia momen (in^4)

The vibration absorber design process begins with determining the material and beam dimensions. In accordance with the design drawings and sizes above, and from equation 4, we get the weight of the concentrated mass w_2. Data needed in the calculation as follow:

Material: carbon steel
Modulus Young (E): 29.000.000 psi
Density: 0.282 lb/in^3 = 7805,733 kg/m^3
L : 500 mm = 19,685 in
b : 40 mm = 1,575 in
h : 20 mm = 0,787 in
a : 350 mm = 13,779 in
N_f : 1500 rpm
I : $\frac{1,575(0,787^3)}{12} = 0,06398$ in^4

w_1= ((19,685 x 1,575 x 0,787)/19,685) in^2 x 0,282 lb/in^3 = (6,883/19,685) lb/in

Then,

$$w_2 = \left(\frac{(2.114x10^5)E.I}{Nf^2.(3a^2L - a^3)}\right) - \frac{0.75 w_1 L^4}{3a^2L - a^3}$$
$$= \left(\frac{2.114x10^5 x 29x10^6 x 0,06398}{1500^2 x (3x13,779^2 x 19,685 - 13,779^3)}\right) - \frac{0,75x(6,883/19,685)x19,686^4}{3x13,779^2 x 19,685 - 13,779^3}$$
$$= 15,68$$

Getting value w_2 = 15,68 lbs (7,1 kgs)

After the value of w_2 is known, the dimensions of the second workpiece using carbon steel plates are determined with the dimension length x width x height, 170mm x 112 mm x 23 mm. The following is a picture of a vibration absorber designed according to the above calculations (Fig.3). Furthermore, the vibration absorber that has been fabricated is mounted on a pump that experiences vibration due to resonance as discussed earlier (Fig. 4).

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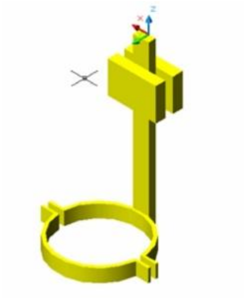

Figure 3: Vibration Absorber Designed

Figure 4: Installation of Vibration Absorber on CEP

III. RESULTS AND DISCUSSIONS

3.1 Experiment on Natural Frequency Shifting

This experiment was carried out to prove that there was a change in the natural frequency for the system with the vibration absorber installed. Experiments were carried out on a system that has a natural frequency of 21.41 Hz. The natural frequency value is obtained by doing a bump test on the system. The position of mass we will be varied at three points conditions, by experiment separately:

a) Condition A: 20 mm from the base
b) Condition B: 250 mm from the base
c) Condition C: 450 mm from the base

Without vibration absorber, the system has a natural frequency 21.41 hz (Fig.5). Natural frequency system with vibration absorber at condition A, there is a shift in the natural frequency of the system from 21.41 Hz to 20.71 Hz (Fig 6). Natural frequency system with vibration absorber at condition B, there is a shift in the natural frequency of the system from 21.41 Hz to 20.24 Hz (Fig 7). Natural frequency system with vibration absorber at condition C, in this condition, there is a shift in the natural frequency of the system from 21.41 Hz to 17.29 Hz, but the second natural frequency of the system approaches the natural frequency of the system without an absorber (Fig 8).

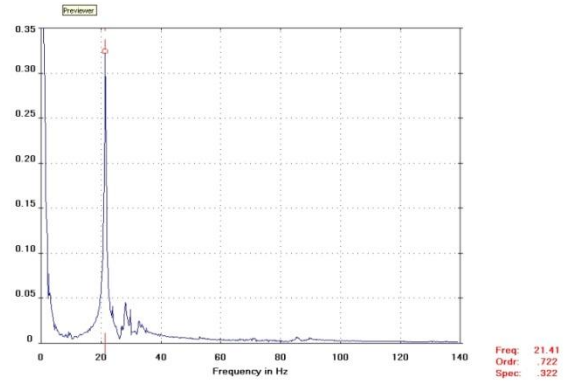

Figure 5: Initial Natural Frequency

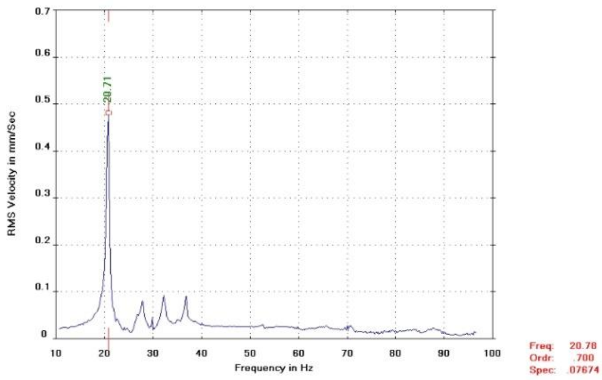

Figure 6: Natural Frequency Condition A

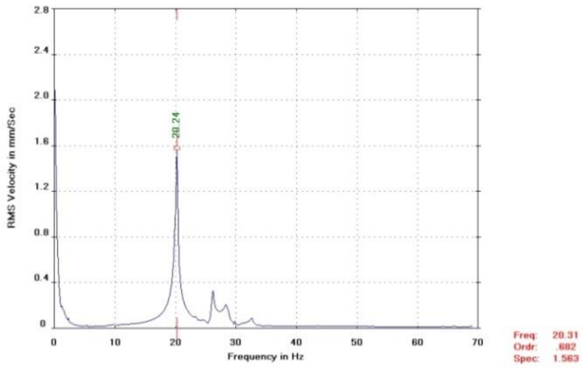

Figure 7: Natural Frequency Condition B

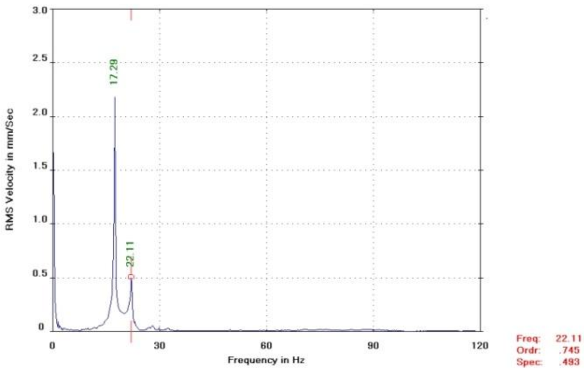

Figure 8: Natural Frequency Condition C

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3.2 Adjusting Position Absorber

As previously explained, foot-related resonance is directional, which means that the amplification of vibrations only occurs in one particular direction. The stick of the vibration absorber has a cross-sectional area of **b** x **h**, where the values of **b** and **h** are not the same. This result in the value of the moment of inertia is not the same in the **b** and **h** directions. This difference will result in different stiffness values which will ultimately affect the natural frequency value of the vibration absorber.

The influence of dimensions on the stiffness of the vibration absorber will affect the position and direction of the vibration absorber installation. Improper positioning will result in suboptimal damping. To get optimal damping results, it is necessary to adjust the position of the mass w. Three times adjusting the position and measurements obtained the following results (Table 1):

Table 1: Data Comparison Vibration Value

	Initial Value	**Tunning I**	**Tunning II**	**Tunning III**
MOH	7	4,5	2,1	1,6
MOV	1,9	2	1,8	1,7
MOA	0,6	0,7	0,8	0,7
MIH	2,5	1,4	0,9	0,7
MIV	1,3	0,9	0,9	0,7
PIH	0,8	0,6	0,5	0,3
PIH	0,6	0,8	0,5	0,5
PIA	0,3	0,6	0,5	0,3

Notes: M (Motor), P (pump), O (onboard), I (inboard), H (horizontal), V (vertical), A (axial). Unit in mm/s.

Based on table 1 above, vibration on the motor outboard horizontal (MOH) was reduced from 7 mm/s to 4.5 mm/s in the first tuning, then dropped again to 2.1 mm/s in the second tuning, and finally to 1.6 mm/s s on the third tuning. Vibration reduction is obtained by about 77% from the initial value.

IV. CONCLUSION

Experiments carried out by conditioning the absorber distance for w2. When tuning the w2 position (while the equipment still operating) shifts are made in the smallest possible increments to get the optimal value, while still measuring vibrations in the main system in order to obtain second natural frequency of the system approaches the natural frequency of the system without an absorber. Installation of the Vibration Absorber on Condensate Extraction Pump can reduce vibration due to resonance up to 77% at the desired frequency.

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